GENERAL MOTORS
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for educators in the fields of engineering and allied sciences

FUEL INJECTION

Some Thoughts on Knowledge

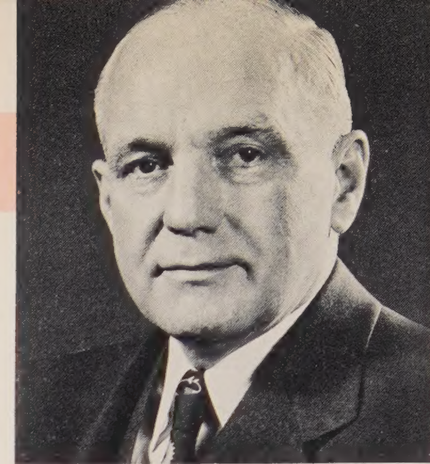
Knowledge is our most important national resource. It is the only sure bulwark of national security—in both a military and an economic sense. But it is a resource that must be developed further if we are to maintain superiority in defense technology and if we are to advance our general economic and cultural level.

That we must advance our economy can be seen from certain definite facts—not hopeful predictions—that affect the growth of the United States. Our population is increasing at a rate that is expected to reach a total of 191 million by 1966. This is a 12 per cent expansion over today's figure of about 170 million. Increasing at an even faster rate than the population growth is the number of households. These pressures mean that we will need to advance our economy merely to hold our own with respect to our present living standard. But it is not our tradition to stand still. We are constantly seeking improvements in all aspects of our lives—better homes, automobiles and medical care; expanded opportunities for educational and cultural development. Thus, we will need to advance our economy still further if we are to accomplish these natural improvements in our standard of living.

These demands make clear the need for the development of scientific and technical knowledge. For when we improve technology, we create an atmosphere favorable to the further development of our society in all of its respects. A people preoccupied with scratching out a bare existence has little time or inclination to devote itself to cultural pursuits. We Americans are sometimes accused of being too materialistic and too little concerned with the nobler things of the mind and spirit. Material progress and a cultural and spiritual growth should not be considered mutually exclusive. In fact, as a nation, we have demonstrated that they are not. While we have improved technology, we also read more good books than ever before; we enjoy more good music and drama; we attend our churches in greater and steadily growing numbers. Hence, it is fair to conclude that rather than being a bar to cultural development, material progress is a prerequisite.

To advance our technology—and thus our economy—it is broadly recognized that we must have the manpower with the requisite mental ability and education. The problems of education and of assuring our further progress in technology are so closely intertwined as to be inseparable.

Our young people need to be imbued with a respect for education as a high privilege. They need to be motivated to develop their God-given abilities to make a worthwhile contribution to society. These are responsibilities that begin in the home but continue in the schools. There are responsibilities falling upon the corporate citizen, too. Industry can be of appropriate help in the field of education in many ways. For example, some representative General Motors helps in this area are the maintaining of close liaison with educators at all levels of education; offering financial aid under several programs to students and to institutions; offering various types of educational aids; and arranging summer conferences, or—in certain cases—related work experience for science and engineering educators.



All of us, whether as citizens, educators, or businessmen must share in the task of expanding our most fundamental national resource—knowledge. If as a society, we would *have* more, we must increase our ability to *produce* more with the same amount of human effort. As our knowledge expands and we learn to produce with greater efficiency, we can look forward with some degree of confidence to the further growth of our economy and to the cultural and spiritual development of our people.

S. E. Skinner,
Vice President and
Group Executive

THE COVER

This issue's cover, designed by artist Ernest W. Scanes, symbolically represents the inner workings of the recently developed General Motors fuel injection system. The swirl pattern represents the flow of air on its way to being mixed with fuel shot from the fuel injection nozzle. Air enters the system through an annular venturi section. The velocity of air flowing through the venturi creates a pressure signal, the intensity of which is proportional to the mass of air flowing. This signal, in turn, acts on the mechanism controlling the amount of fuel injected by the injection nozzle into the intake port behind the inlet valve of each cylinder. The end result is a correct air-fuel ratio admitted to the combustion chamber for all driving conditions.

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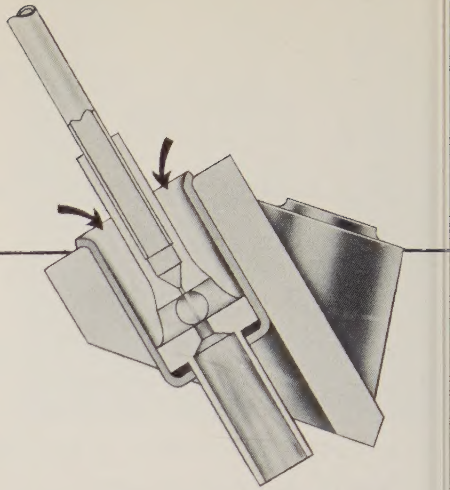
A Discussion of the Basic Design and Operation of the General Motors Fuel Injection System



A fuel injection system for spark ignition engines which would successfully overcome certain inherent design problems associated with the conventional carburetor has been a longtime objective of automotive engineers. The many advantages to be gained from fuel injection, such as faster cold weather starting and warm up, better fuel economy, and instantaneous response to throttle movements, have long been known. In addition, fuel injection also permits the use of dynamic supercharging which results in increased power and torque output. The design of a fuel injection system for a spark ignition engine, however, involves much more than merely replacing the carburetor by an injection nozzle and pump. In addition to overcoming the design problems inherent in a carburetor-manifold combination, the injection system must be designed to provide the advantages possible. It also must be able to compete with the carburetor for low cost, simplicity, reliability, and low-speed performance. The majority of injection systems developed have usually been too complicated and costly for widespread application to automotive engines. A few years ago, the General Motors Engineering Staff (located at the GM Technical Center) initiated a program having as its objective the design and development of a fuel injection system which would be simple in design, reliable in operation, and reasonable in cost for application to automotive engines. The result has been a fuel injection system, based on the continuous flow principle, which has been adapted by Chevrolet Motor Division engineers for application on the Chevrolet Corvette and stock passenger cars as optional equipment. Pontiac Motor Division also has adapted the basic system for use on its Bonneville sports convertible.

The carburetor used on today's automotive engines has three basic functions: (a) it meters fuel to satisfy all engine conditions such as speed, load, temperature, altitude, and operating throttle transients, (b) it mixes air and fuel uniformly for proper combustion, and (c) it transports the air-fuel mixture to the intake manifold for distribution to the individual cylinder intake ports of the engine.

Through the years, the carburetor-manifold has performed satisfactorily its role in the fuel distribution system of the automobile. The average carburetor-manifold, however, has certain inherent

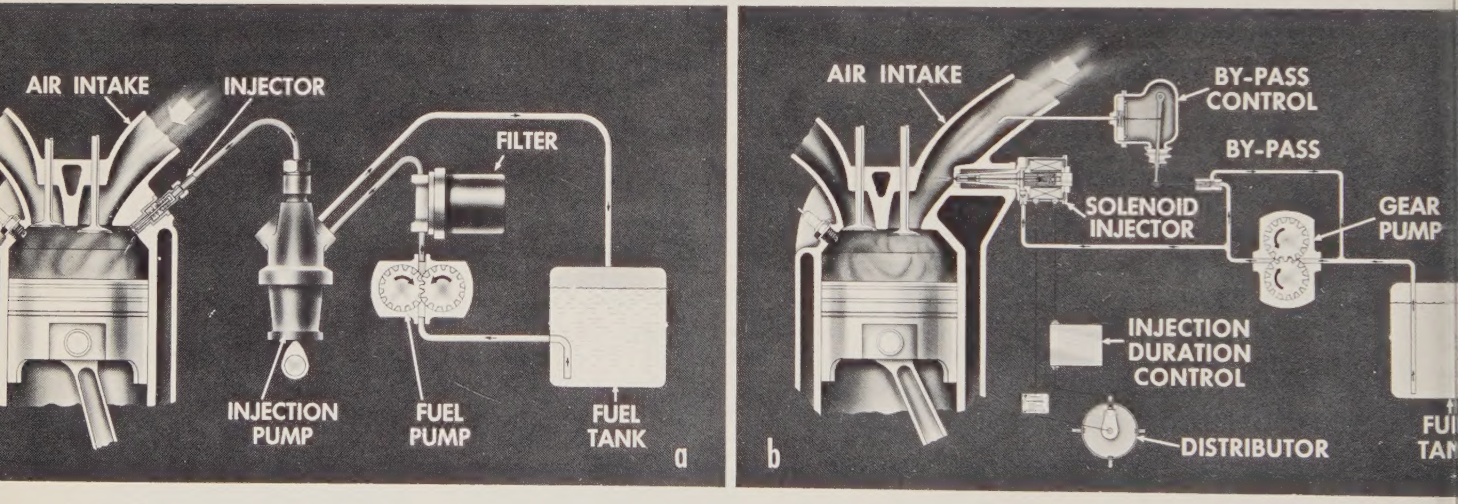


Fig. 1—Various types of existing fuel injection systems used on spark ignition engines were studied during the early developmental stage of the GM continuous flow fuel injection system. One of the systems studied was that used on an aircraft engine (a) in which fuel was injected directly into the combustion chamber by a Diesel-type injection pump provided with metering controls. This system did a good job of fuel distribution in the rather narrow cruising and full power range of aircraft engines. The injection system not only overcame the difficult fuel distribution problem on radial air-cooled aircraft engines, but also eliminated the fire and explosion hazard created by the combustible mixture in the manifold and supercharger of carbureted radial and V-type engines. Analysis of the aircraft type of fuel injection system indicated that the design of the nozzle could be simplified and its reliability enhanced by injecting fuel into the intake ports rather than into the combustion chamber. Also, the use of individual plungers for each cylinder was desirable for aircraft engines, which operate at fairly substantial bmep, but was undesirable for automotive engines operating under idle and city driving conditions because of poor distribution characteristics inherent to the system at low fuel supply rates. Another fuel injection system studied was the Fuscaldo system (b). This system consisted of a source of fuel under pressure at the engine intake ports. Electromagnets opened precision-made valves at each intake port to deliver fuel in relation to engine requirements. While the Fuscaldo system offered some simplification over the Diesel-type pump and injector used on the aircraft injection system, it was considered still too complex for automotive engine application.

By JOHN DOLZA
General Motors
Engineering Staff

One approach to utilizing the full power and work capacity of an automotive engine

limitations that necessitate a design which constitutes an optimum compromise to the metering, mixing, and distribution functions required. Engineers have felt that if some of the limitations inherent with the carburetor-manifold could be eliminated, greater improvements in fuel economy and power output could be realized, in addition to easier starting and faster engine warm up in cold weather and quicker response to the throttle during operating transients.

It was with this thought in mind that the GM Engineering Staff initiated a design and development program to achieve a fuel injection system for spark ignition engines which would provide a solution to the design problems inherent with the carburetor-manifold, be simple in design and reliable in operation, and be economical to manufacture.

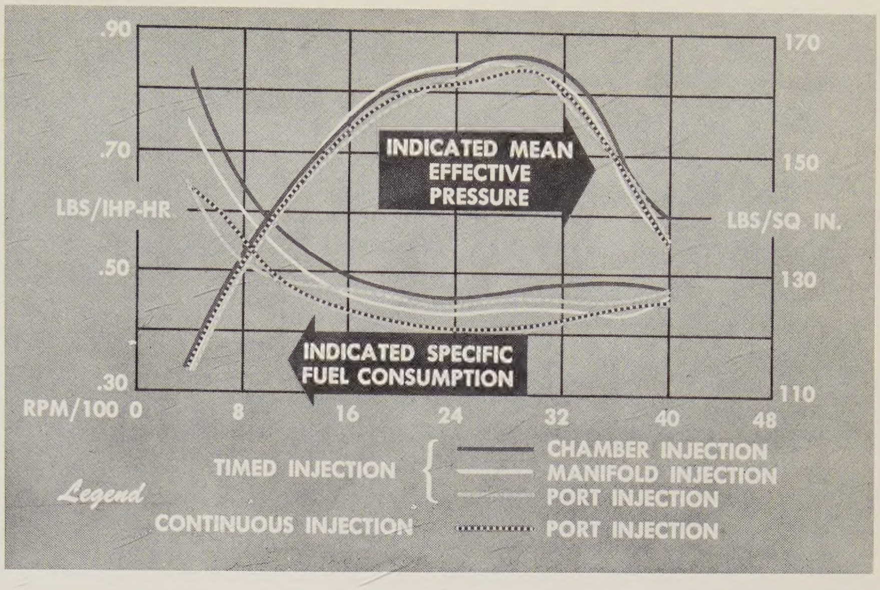


Fig. 2—Basic developmental test work on the GM fuel injection system was performed on a single cylinder engine having a compression ratio of 8 to 1. Tests were run to compare timed and continuous injection. The upper set of curves shows the indicated mean effective pressure (lb per sq in.) obtained at various engine speeds with timed and continuous injection. The lower set of curves shows the indicated specific fuel consumption (lb per ihp-hr). The test results established three points: (a) direct injection into the combustion chamber had no appreciable advantage over injection into the intake ports, (b) intermittent, or timed, injection into the intake ports gave slightly less power and used slightly more fuel than continuous injection, and (c) injection directed toward the intake valve gave the most power, the best economy, the fastest warm up, and the best acceleration response. Based on these facts the GM Engineering Staff chose continuous flow injection into the intake ports with open orifice nozzles directing the fuel toward the intake valves.

Existing Fuel Injection Systems Studied First

The early phases of the developmental program were devoted to the study and evaluation of existing fuel injection systems used on spark ignition engines. Typical of the systems studied were an aircraft engine type of fuel injection system (Fig. 1a) and a system of Italian design—the Fuscaldo system (Fig. 1b).

A detailed study of the aircraft engine type of injection system indicated that because of inherent fuel distribution characteristics at low fuel supply rates the system would not be suitable for automotive engines when operating under idle and city driving conditions. In addition, the system would be too high priced for automotive use, even with volume production. The Fuscaldo system offered some simplification over the aircraft-type of injection system, but was

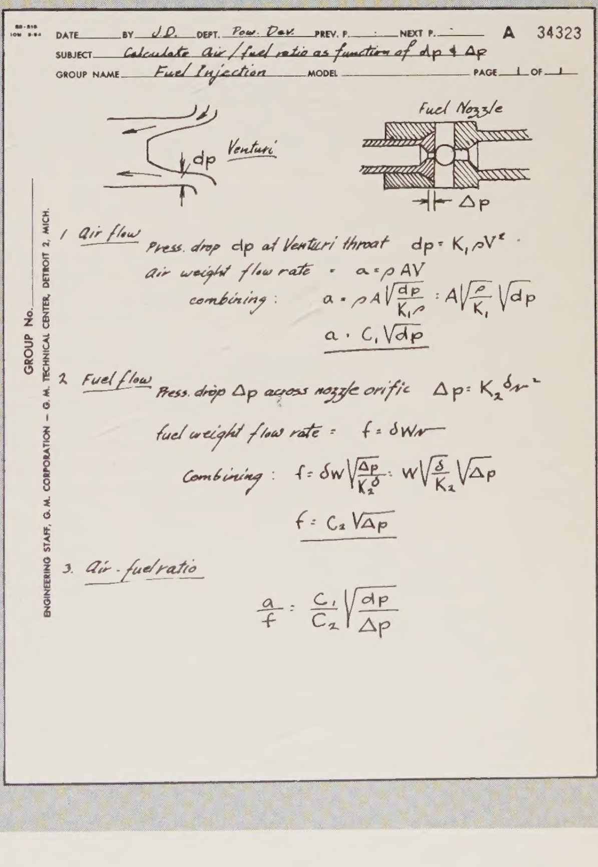


Fig. 3—The air fuel ratio for the fuel injection system was calculated as a function of dp, the pressure drop at the venturi throat, and Δp, the pressure drop across the orifice of the fuel nozzle. In the calculations for air flow, K_1 = the venturi coefficient, ρ = the air density, V = the air velocity, A = the venturi throat area, and $C_1 = A\sqrt{\rho/K_1}$. In the calculations for fuel flow, K_2 = the fuel nozzle orifice coefficient, δ = the fuel density, v = the fuel velocity, W = the fuel nozzle orifice area, and $C_2 = W\sqrt{\delta/K_2}$. The derived expression for the air-fuel ratio shows that the air-fuel ratio, by weight, is proportional to the square root of the ratio of air pressure drop at the venturi throat (venturi pressure) to fuel pressure at the nozzle orifice (fuel pressure).

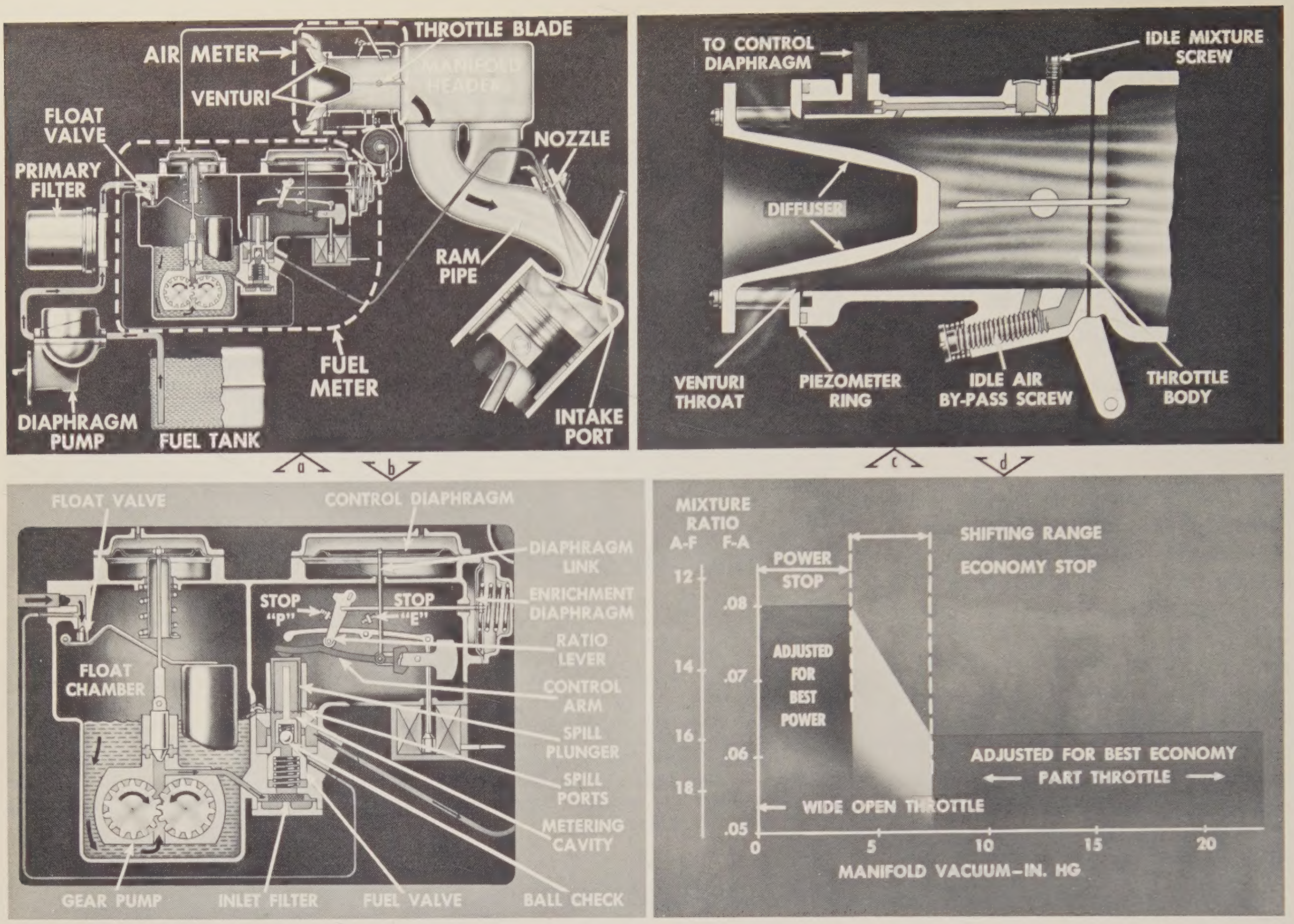


Fig. 4—Two major components of the basic GM continuous flow fuel injection system (a) are the fuel meter and the air meter. Fuel supplied by a conventional diaphragm pump passes through a primary filter before entering the fuel meter. The fuel enters the fuel meter through a float valve, similar to its counterpart in a carburetor, and then passes into a float chamber (b) where any vapors formed because of temperature rise are vented. A gear pump at the bottom of the float chamber delivers the fuel through a second inlet filter and a fuel valve to the metering cavity. The fuel valve contains an antipercolation ball check to keep the fuel, between the valve and the gear pump, at a pressure sufficiently high to eliminate vapor pockets. Some of the fuel delivered to the metering cavity flows directly to the continuous-spray nozzles. The remaining fuel flows through spill ports back to the float chamber. The amount of fuel spill is regulated by the spill plunger.

Air enters the air meter (c) through the annular venturi section and flows past the throttle blade to the manifold header and then to the individual intake ports. The velocity of air flowing through the venturi causes a depression signal related to the mass of air flow. This venturi signal acts on the control diaphragm of the fuel meter to create a force which is transmitted to the spill plunger by means of the diaphragm link and control arm, the latter being pivoted on the ratio lever (b). An increase in air flow through the venturi causes a relative increase in venturi signal which, acting on the control diaphragm, results in an increase in the force acting on the top of the spill plunger. The spill plunger then moves to a new balanced position to obtain a fuel pressure increase proportional to the venturi signal increase. Since the increased fuel pressure results in a fuel flow proportional to the increase in air flow indicated by the venturi signal, a constant air-fuel ratio is maintained as long as the ratio linkage is not changed.

The air-fuel ratio can be varied by changing the linkage leverage between the control diaphragm and the spill plunger. This is accomplished by shifting the pivot point on the control arm. When the ratio lever rests against stop *P* (power) the air-fuel ratio obtained is for wide-open throttle operation. When the ratio lever rests against stop *E* (economy) part throttle fuel requirements are obtained. For automatic operation, the ratio lever is controlled by a spring-loaded enrichment diaphragm which is subjected to manifold vacuum. At light load (high vacuum) the lever is held at position *E*. At full load (low vacuum) the lever is held at position *P*. The stops are set to obtain a best economy air-fuel ratio (about 15.5 to 1) at part throttle and a maximum power air-fuel ratio (about 12.5 to 1) at wide-open throttle (d). The enrichment diaphragm spring is set to hold the ratio lever in the economy position above about seven inches of mercury vacuum.

considered still too complex for automotive engine application.

A thorough analysis was made of other existing fuel injection systems. It became evident, however, that none of the existing systems could compete with the carburetor-manifold combination for simplicity, low cost, reliability, and low-speed performance and at the same time provide a solution to the design problems inherent in the carburetor-manifold combination. In view of this, the GM Engineering Staff decided to start from the beginning and design and develop its own fuel injection system.

Basic Developmental Work Performed on Single Cylinder Engine

The basic developmental test work on the GM fuel injection system was done on a single cylinder engine having a compression ratio of 8 to 1. Based on facts obtained from test results (Fig. 2) a continuous flow fuel injection system was chosen in which fuel would be delivered into the intake ports. Open orifice injection nozzles would direct the fuel toward the intake valves.

After the type of fuel injection system was chosen, developmental work was

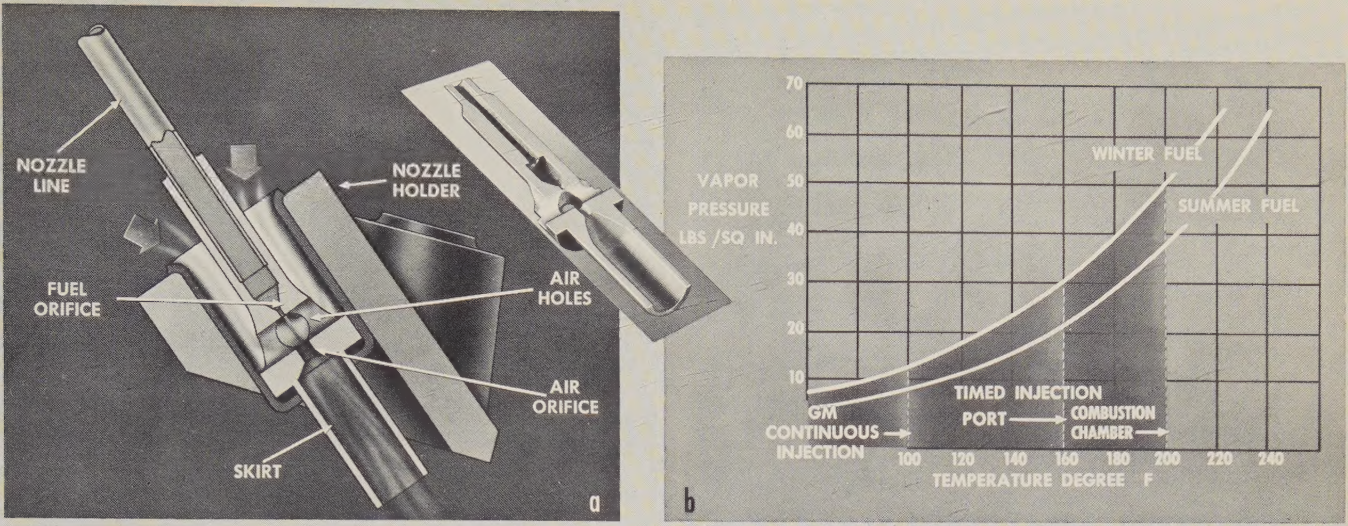


Fig. 5—The fuel injection nozzle (a) is mounted in a plastic holder at the entrance to each cylinder intake port. Each nozzle has an accurately calibrated open fuel orifice slightly less than 1/64 in. in diameter. Just below the fuel orifice is a small air chamber which receives filtered air through four 0.100-in. diameter holes from the air cleaner. This air supply assures that the nozzle discharge is at all times near atmospheric pressure regardless of manifold vacuum fluctuations. The amount of fuel injected, therefore, depends solely on the metering system pressure. Directly in line with the fuel orifice and across from the air chamber is a 0.040-in. diameter air orifice opening to the cylinder intake port. Under all manifold vacuum conditions the atomization of fuel is enhanced by the mixing of air and fuel through the air orifice. The volume of air passed through the air orifice represents about one fourth the air used at closed throttle for idling operation. Below the air orifice is a tubular skirt which is cooled by evaporation of the fuel. The nozzle fuel line, in turn, is cooled by conduction. This cooling prevents metering disturbances caused by fuel vapor bubbles. The fuel at the nozzle and in most of the nozzle line is at a temperature lower than the temperature it had when leaving the diaphragm pump. This method of refrigerating the fuel by means of evaporation allows the open orifice type of nozzle to handle high vapor pressure fuel even under summer operating conditions. The low temperature operating range of the nozzle (b), due to its location and insulation, greatly reduces vapor problems.

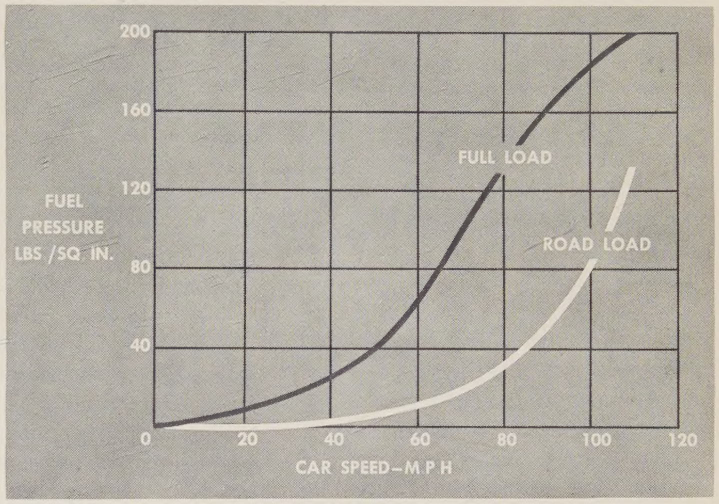


Fig. 6—Under normal driving conditions fuel pump pressure seldom exceeds 20 psi. For cranking and idling operation, the fuel pressure is approximately eight psi. A maximum pressure of approximately 200 psi is available during top engine speed and wide-open throttle operation.

then undertaken on every component to be used in the system. The developmental work represented successive steps in simplifying the design, improving the method used for metering the fuel and air, and increasing the overall reliability of the system.

Two Methods of Air and Fuel Metering Considered

Two methods for metering the air and fuel were considered for the continuous flow fuel injection system: (a) engine speed-air density metering method and (b) mass flow metering method.

Engine Speed-Air Density Metering Method

The engine speed-air density metering method is used to control aircraft engine fuel injection systems. The use of this method may have been dictated by the wide range of air density encountered during flight conditions.

The sensor of pressure and temperature used with this method of metering fuel and air must correct in proportion to absolute pressure and absolute temperature. For example, an ambient temperature change during flight of from $-30°$ F to $+100°$ F, or 430° R to 560° R absolute temperature, would require a metering change of 560/430 or 30 per cent. This means that air at $-30°$ F requires 30 per cent more fuel than the same volume of air would require at $+100°$ F. A like change would be shown for differences in atmospheric pressure, similar to what an automotive engine might encounter when climbing Pike's Peak.

The engine speed-air density method of metering needs a device which will meet two requirements: (a) sense absolute temperature and absolute pressure accurately and (b) correlate these absolute values to the metering means. These two requirements make the sensing device costly. Also, this type of metering method requires a means to measure engine volumetric efficiency versus engine speed which, on "rammed engines," would be rather complicated and costly. In addition, the engine speed-air density metering method usually requires the

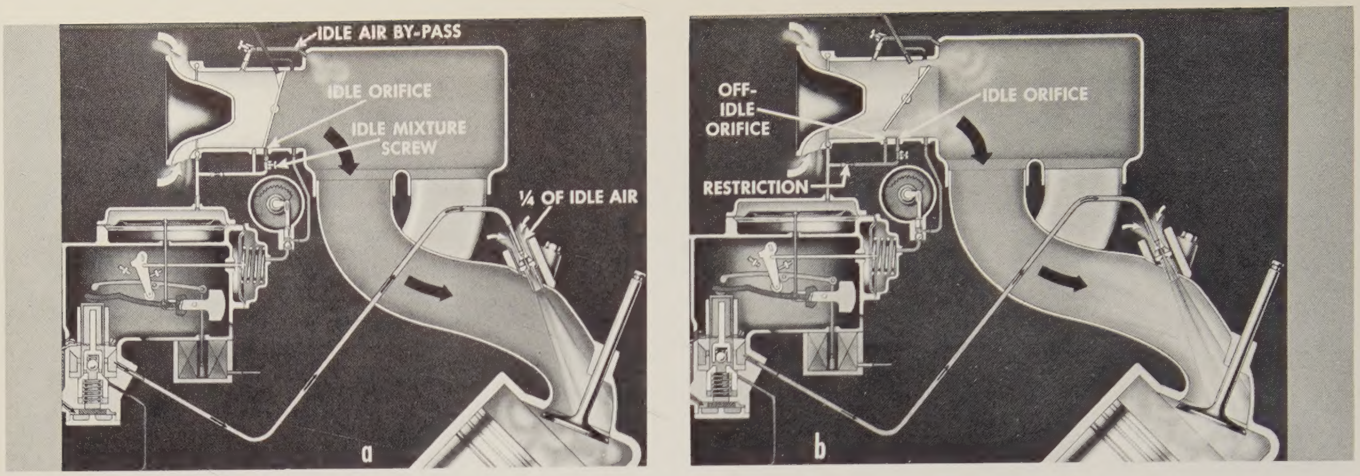


Fig. 7—During idle operation the throttle is in a closed position (a). About ¼ of the air required for combustion is supplied through individual nozzle air chambers located just below the fuel orifice. The remaining air is supplied through the idle air by-pass. This air is regulated by an idle air by-pass adjusting screw located within the throttle body. The correct mixture for idling is set by means of the idle mixture screw. This screw controls the amount of venturi signal boost caused by bleeding in manifold vacuum from the idle orifice. For best economy during steady state operation at idle and off-idle, the engine requires a richer mixture than that required at higher speeds. Part of the enrichment is supplied by the basic design and placement of the spill plunger and nozzle. The remainder of the enrichment required at idle is supplied through the idle orifice and the remainder required in the off-idle range is supplied through the off-idle orifice (b). As the throttle is opened from the idle position, the off-idle orifice is subjected to manifold vacuum. This serves to intensify the venturi signal. The amount of off-idle bleed is regulated by a restriction in the bleed passage.

same amount of fuel per cycle. This requires, in turn, the use of a uniform flow fuel pump which must not depreciate in service.

Mass Flow Metering Method

The mass flow metering method operates as a function of the mass of air consumed by the engine—in other words, a method which uses a high efficiency venturi. With this type of metering method, the temperature and pressure effect is corrected only as a function of the square root. For example, an absolute temperature change of from 460° R to 560° R would require a metering change of $\sqrt{560/430}$ or 14 per cent, compared to 30 percent for the engine speed-air density metering method operating under the same temperature change. Out of the 14 per cent metering change required, fuel density variations with temperature would cancel four per cent. Proper utilization of fuel vapor bubbles at high temperatures would tend to cancel a large part of the remaining 10 per cent.

With the mass flow metering method the volumetric efficiency of the engine is sensed by the venturi. No additional correcting signal is required. Also, constant fuel delivery is not required, as with the engine speed-air density method, as long as delivery from the fuel pump is in excess of the desired flow at the nozzles.

Air and Fuel Metered by the Mass Flow Method

The method selected for metering air and fuel for the GM fuel injection system is based on the mass flow metering method in which venturi throat depression is related mechanically to fuel pressure. A mathematical derivation for the air-fuel ratio by weight (Fig. 3) showed that the ratio is proportional to the square root of the ratio of venturi pressure to fuel pressure.

Two major components of the GM fuel injection system (Fig. 4a) are the fuel meter (Fig. 4b) and air meter (Fig. 4c). The fuel meter is designed with a linkage leverage arrangement which allows a variation in air-fuel ratio corresponding to throttle requirements. Best fuel economy at part throttle is obtained with an air-fuel ratio of about 15.5 to 1. A maximum power air-fuel ratio of about 12.5 to 1 is obtained at wide open throttle (Fig. 4d).

The fuel injection nozzles (Fig. 5a) used with the system have an accurately calibrated open fuel orifice. Air supplied to a point immediately below the fuel orifice assures that the nozzle discharge is at all times near atmospheric pressure regardless of manifold vacuum fluctuations. As a result, the amount of fuel injected depends solely on the pressure of the metering system.

The fuel injection nozzle is so designed that the fuel is refrigerated by means of evaporation. The operating temperature range of the nozzle (Fig. 5b) is such as to greatly reduce vapor problems.

The fuel orifice is designed to give a minimum pressure of about eight inches fuel head during cranking and idling operation and a maximum pressure of about 200 psi at top engine speed and wide-open throttle operation. Under normal driving conditions the pump pressure seldom exceeds 20 psi (Fig. 6). These moderate pressures are easily achieved with a plain gear pump. The fuel pressure at cranking and idling has been found sufficiently high to prevent adverse fuel distribution effects caused by normal engine attitude changes.

When the engine operates under idling conditions about ¼ of the air required for combustion is supplied through the individual nozzle air chambers located just below the fuel orifice. The remaining air is supplied through a passage around the throttle blade and is regulated by means of an adjusting screw (Fig. 7a).

For best economy during steady-state operation at idle and off-idle, the engine requires a mixture which is richer than that required at higher speeds. The basic venturi system results in a leaner mixture at low rates of air flow but, because of the weight of the spill plunger and the location of the injection nozzle below the spill level in the fuel meter, there is some inherent fuel enrichment near idle. The remainder of the required idle mixture enrichment is supplied through an idle orifice and the remainder of the enrichment required in the off-idle range is supplied through an off-idle orifice (Fig. 7b).

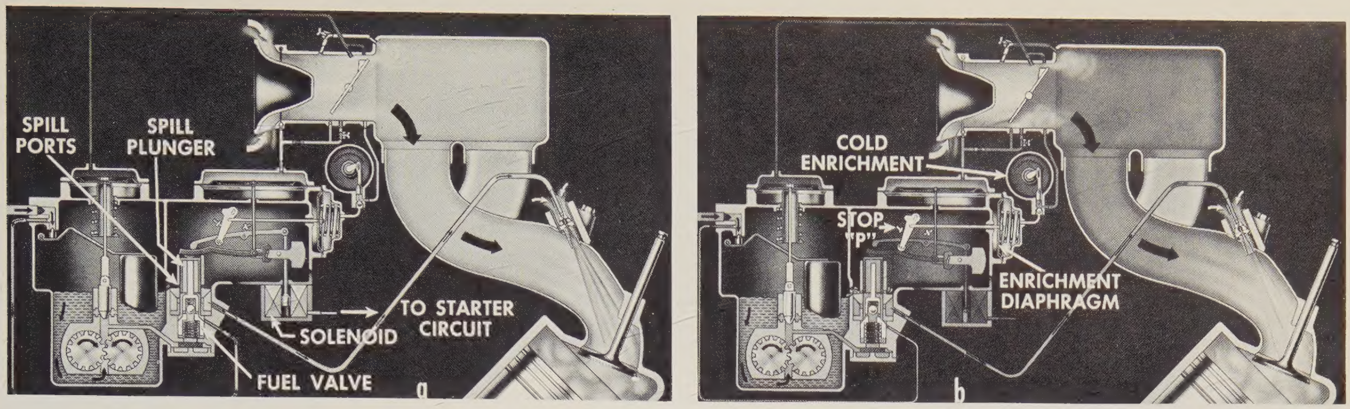


Fig. 8—During cold start operation a certain amount of fuel enrichment is required. The enrichment required is achieved by the action of a solenoid (a), connected to the starter circuit, which forces the spill plunger down far enough to unseat the fuel valve. This action cuts off completely the spill by covering the spill ports while at the same time uncovering a passage from the float chamber inlet. This allows extra fuel to be supplied to the injection nozzle directly from the diaphragm pump. While the engine warms up, a cold enrichment mechanism (b) keeps the air-fuel ratio lever at the stop *P* (power) position by cutting off manifold vacuum to the enrichment diaphragm. This enrichment assures steady-state engine operation during warm up. Because the fuel stream is directed upon the intake valve, the incoming mixture is warmed at a rate considerably faster than would be the case in a carbureted engine having a "hot spot" manifold. The engine idle speed is kept high enough to eliminate stalling by means of a fast-idle cam which opens the throttle blade slightly during warm up.

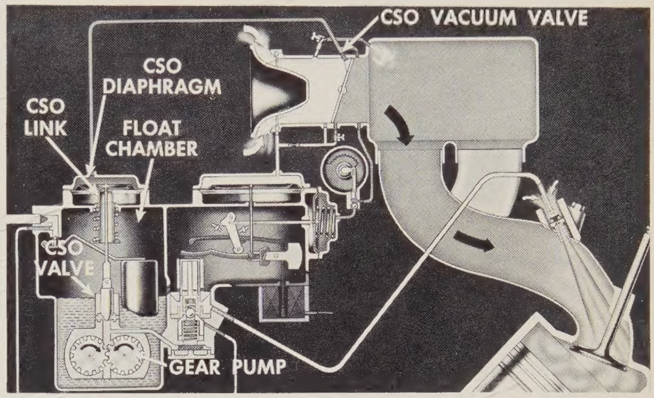


Fig. 9—Exhaust fumes containing unburned fuel are eliminated during coasting by means of a coasting shut-off (CSO) valve and linkage arrangement located directly above the gear-pump outlet. The CSO consists of a valve which, under normal operating conditions, is closed by a spring. A CSO link connects the valve to the CSO diaphragm which is subjected to manifold vacuum at the throttle body. During coasting, high manifold vacuum exerts a force on the diaphragm in excess of the spring load. The pull rod then lifts the valve off its seat, thereby allowing the pump to discharge fuel directly back to the float chamber instead of to the injection nozzles. The vacuum valve is so arranged that vacuum can be applied to the diaphragm only when the throttle is closed to eliminate CSO action as the engine is accelerated at no load.

Design Eliminates Need for Accelerator Pump

The GM fuel injection system does not require an accelerator pump to supply extra fuel during throttle opening transients for two reasons: (a) manifold wetting is kept to a minimum and (b) there is instantaneous fuel meter response. When the throttle opens, an inrush of air immediately fills the manifold. This inrush of air is greater than the stabilized air requirement and causes an equivalently greater venturi signal. Since the linkage parts of the fuel meter are very light and the travel small, their inertia is negligible. The inrush of air, therefore, causes an instantaneous rise in fuel pressure. The result is a shot of fuel proportional to the inrush of air.

Cold-Start Fuel Enrichment Achieved by Solenoid Action

In a carbureted engine, fuel enrichment is required to wet and flush the intake manifold during cold-start operation. For the fuel injection system, substantial fuel enrichment is not required because fuel is distributed directly to the intake ports. The fuel enrichment that is required for cold starting is achieved by means of a solenoid connected to the starter circuit (Fig. 8a). While the engine warms up, a cold-enrichment mechanism (Fig. 8b) keeps the air-fuel ratio lever at the position which assures steady-state engine operation during warm up.

Coasting Shut-off Valve Eliminates Unburnt Fuel in Exhaust

Because the fuel injection system operates with a dry manifold, the amount of fuel vaporization in the manifold during deceleration at closed throttle is negligible. To eliminate exhaust fumes containing unburnt fuel during coasting, a coasting shut-off valve is provided (Fig. 9) which allows the fuel pump to discharge fuel, during coasting, directly back to the float chamber instead of to the injection nozzles.

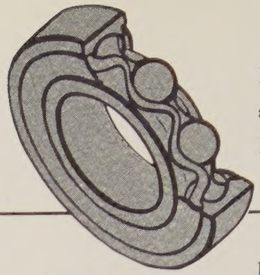
Conclusion

Fuel injection systems for spark ignition automotive engines differ from each other both in basic operating principles and in the design of system components. The GM fuel injection system represents but one approach to providing the means for utilizing fully the power potential and work capacity of the engine.

The basic design of the fuel injection system developed by the GM Engineering Staff has been adapted by Chevrolet engineers to the Chevrolet Corvette and stock passenger cars. The results have shown that the theoretical advantages offered by fuel injection are a reality.

As is the case with all new engineering developments, the fuel injection system will continually be improved upon. The need for fuel injection has been established. What the future will bring will depend on the abilities of engineers to simplify still further the overall design and develop new or better ways for economical manufacture.

Properties of Integral Seal Ball Bearings and Their Use by the Engineer



By MICHAEL T. MONICH
and CARTER T. BRAGDON
New Departure Division

Integral seal ball bearings can simplify product design if properly selected and applied

The expanding use of "lubricated-for-life" integral seal ball bearings has resulted from the important savings and product improvements which they have made possible. To take advantage of these improvements, the designer needs to be acquainted with the capabilities and limitations of each basic type of integral seal. A comparative seal evaluation, based on the standardized laboratory procedures employed by New Departure Division in evaluating seal and seal materials, is helpful for this purpose. New Departure also has furnished information concerning the effects which operational factors, such as temperature and speed, have on the grease life of sealed bearings. A study of the actual applications reveals the many different ways in which integral seal bearings are employed, and two factors stand out: design simplification and absence of maintenance requirements. In preparation for future needs, current trends must be observed and employed to guide future seal development.

Early designs of ball bearings were generally of the open type, that is, they consisted primarily of the inner and outer ring between which a separator and a complement of balls were placed. As the realization grew that ball bearing life was affected by exposure to abrasive foreign matter, various external closures were devised which were held by either the shaft or housing member to act as a slinger or baffle, or as a friction seal having rubbing contact with the rotating member. The integral seal came as a later development, resulting from design demands for a "package" unit consisting of a ball bearing having integral seals which would be permanently grease lubricated for life by the ball bearing manufacturer.

Integral seals, as the name implies, are built-in seals which are part of the ball bearing itself. While some seal designs require that the bearing width be extended to accommodate them, others are incorporated into the standard bearing widths established for open bearings.

The primary purposes of the integral seal are similar to those of separate seals; namely, they are intended to keep contaminants out and the lubricant in the bearing. The secondary purposes, however, are considerably different; namely, they aid in the design of the overall mechanism by simplification and they decrease, and often eliminate entirely, the bearing maintenance on a machine or device. When the aforementioned primary objectives are easily met, the secondary objectives become all-important since they generally result in a more efficient, as well as a less costly, design.

Operating factors of the bearing application, especially speed, temperature, type of contaminant present, and type of lubricant, dictate the seal design and sealing material which will function most efficiently.

Basic Types of Seals are Contact and Clearance Types

Since no single design of a ball bearing seal is equally efficient for all operating conditions, various designs have been developed to meet the requirements of particular cases.

Integral-type ball bearing closures may be divided into two distinct classes: the *contact* type and the *clearance* type. Seals in the contact category consist of those designs in which there is a definite contact pressure between the stationary and rotating parts (Fig. 1). On the other hand, the non-contacting type is designed so that there is a suitable running clearance between the stationary and rotating parts (Fig. 2).

Contact Seals

The object of all contact, or rubbing, type seals is to provide a light contact pressure between stationary and rotating parts but still give sufficient lubricant retention and contaminant exclusion. Control of contact pressure is important since it determines the amount of sliding friction which, in turn, governs the maximum speed which can be obtained without excessive heating. Even with the softest seal material available, too much pressure can result in a damaged or charred seal, or in a scored or burned bearing ring. In addition to control of contact pressure, rubbing seals require a smooth moving surface to minimize wear and friction. The smooth contact surface also is necessary to maintain constant seal drags. To further assure satisfactory functioning and length of service, it is important that the rotating surface be as free as possible of secondary motions, such as eccentricity and face runout between the stationary and rotating parts.

Clearance Seals

Clearance or non-contacting type closures are designed primarily for high speed use where minimum friction is required. This group also may have such supplementary seal devices as slingers or deflectors which aid the primary seal. The simplest form of a clearance seal is the plain type (Fig. 2a). This seal design consists of only one member fixed in the bearing outer ring with a minimum clearance between the seal bore and the outside diameter of the inner ring. It is not a true seal and is generally referred to as a shield or plate. Metal shields are employed to assure an adequate supply of grease in the bearing and are effective in excluding coarse grit or metal chips. Shields also are valuable for keeping dirt out while bearings are being handled or mounted in a machine.

Other clearance-type seals are referred

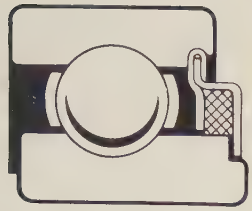


Fig. 1a—*Felt Rubbing-Type Seal:* This is a general purpose seal which meets the requirements of most applications. A felt ring is positioned by two fixed metal members and sealing is accomplished by rubbing contact of the felt washer bore with the smoothly ground OD surface of the bearing inner ring. Control of felt washer dimensions can provide for tight fitting contact for efficient sealing or loose fitting contact for low torque or high-speed operation.

Fig. 1b—*Felt Rubbing Labyrinth Seal:* Although similar to Fig. 1a an additional metal slinger is employed which positions the felt washer for wiping contact with the outer metal shield. Sealing efficiency is thus enhanced. Due to its tight fitup and higher torque, it is suitable primarily for low-speed operation.

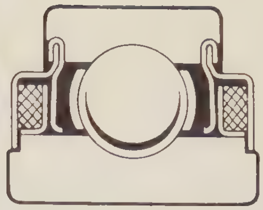
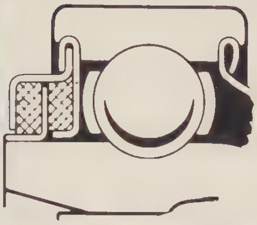


Fig. 1c—*Double Felt Rubbing Seal:* Ability to function under extremely severe operating conditions is possible with this construction which employs both the felt rubbing and labyrinth principles. Additional bearing width is usually required. A tight fitup is usually necessary. This feature precludes its use for high-speed applications.



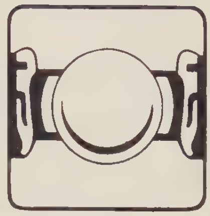


Fig. 1d—*Removable, Notch Rubbing-Type Seal:* This is a highly efficient, universal purpose design employing a molded synthetic rubber sealing member held in position by engaging a groove in the bearing outer ring. Sealing is accomplished by light contact pressure between the seal bore lip and a ground recess on the inner ring of the bearing. When necessary, this seal can be removed for bearing cleaning or relubrication.

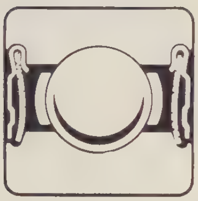


Fig. 1e—*Fixed, Notch Rubbing-Type Seal:* This is a general purpose, narrow width seal similar in principle to Fig. 1d except that the sealing member is generally a stamped rubber diaphragm held in place by a metal shield member. The speed and temperature limits of this seal are governed by the properties of the synthetic rubbing washer.

Fig. 1f—*Spring Lip-Type Seal:* This seal is suitable for bearings exposed to severe drying or fluid contaminating conditions. This design also is effective in retarding oil flow, since the radial pressure exerted between seal lip upon the smooth OD surface of the inner ring provides an effective barrier. Heavy seal pressure precludes use for high speeds.

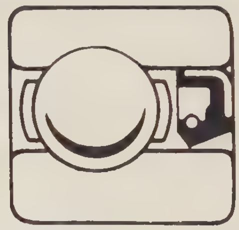


Fig. 1—*Contact*-type seals, as the name implies, are designed so that there is a definite pressure between the stationary and rotating parts of the ball bearing. As explained above, there are a number of arrangements used to meet different design conditions.

to as labyrinth closures (Fig. 2b). With these designs, radial clearances need to be carefully designed to avoid interference, and centrifugal pumping action by the revolving member.

Labyrinth seals are effective for sealing against abrasive type contaminants, but as with the plain type of closure, they are not recommended for sealing against fluids.

Seal Materials Usually Are Felt or Synthetic Rubber

Felt has been used for many years as a sealing element for integral-type closures. Today, in its improved forms, it is still one of the most satisfactory materials for this purpose.

The properties of felt which make it particularly useful as a sealing element are resiliency, low friction, wear resistance, and wiping action of the fibers. In addition to these, it has the ability to absorb lubricant readily due to its porosity. Although this action of oil absorption by the felt provides lubricant at the contact surface, it renders the seal inefficient against gases and liquids unless it is specially treated. Felt material is impervious to the chemical action of most ball bearing lubricants and will withstand temperatures up to 275° F before charring occurs.

Felt may be obtained in two principal forms—woven and pressed. Due to its construction, woven felt is inherently strong and is resistant to permanent distortion. Pressed felt, on the other hand, has poor dimensional control and requires extreme caution in handling. Both types can be used successfully. Due to its superior dimensional control, however, woven felt is generally preferred for integral ball bearing seals.

Use of synthetic rubber for integral-type seals came as a later development and has resulted in a very versatile and useful line of seals. The properties of this material which make it a useful sealing medium are as follows: non-porous, low friction, low permanent set, non-corrosive, non-abrasive, and minimum volume change in most ball bearing lubricants.

Another important feature of synthetic rubber is its ability to be precision molded in small cross sections. This makes it directly applicable to ball bearing use since there is generally a limited space for a seal within a bearing. Synthetic rubber can be bonded to metal. This property not only allows the use of metal inserts to control seal contact pressure but also gives it sufficient retention in the bearing ring. In addition, when seals are fabricated from synthetic rubber (Fig. 1d), they cause negligible OD out-of-roundness at assembly. This is extremely important in bearings requiring precision tolerances.

The synthetic rubber most commonly used in fabricating integral seals is Buna-N. This material can withstand temperatures up to 225° F for continuous operation when exposed to diester, silicone, and petroleum type lubricants. For higher temperature operation, a special polyacrylic type synthetic rubber is available which will operate continuously at temperatures up to 350° F; however, it has a shortcoming in that it deteriorates rapidly when exposed to diester type lubricants at this increased temperature.

Integral Seal Efficiency Evaluated by Test and Field Results

A comparison of the various specific advantages and limitations common to a particular seal design is useful to the designer when considering bearing applications. To establish this information, New Departure developed a bench-test method for evaluating seal efficiency (Fig. 3). This method proved equally effective for checking seals against both fluid and dry contaminants as well as actual grease leakage from a test bearing.

The seal test apparatus consists of a shaft and a U-block support on which test bearings are mounted. Adjacent to the bearing, an agitator paddle rotates in a closed chamber of fixed dimensions to keep the contaminant continually impinging against the seal. The apparatus also permits fluid tests wherein fluids of various viscosities are loaded into stand pipes with fluid heads up to eight inches acting against the bearing seal face. Both of these tests are necessarily accelerated and are not designed to forecast lives of seals or sealed bearings under all applications. The tests, instead, give a comparison of various seals tested under definite controlled types of difficult sealing conditions.

The other requirement of an integral seal is to retain lubricant within the bearing. One method of determining its ability in this respect is to run a double-sealed bearing for a prescribed length of time and at a set speed. Initial and final weight measurements are taken, noting that the external surfaces of the bearing are dry in both cases. Grease leakage is the difference between weights and is expressed as a percentage of the original amount of grease in the bearing.

From these laboratory tests, and from

Fig. 2a—*Shield-Type Closures:* Thi design incorporates a simple metal or plastic dis fitted to the bearing outer ring with running clearance with the inner ring. Although low in efficiency, when compared to the types previously described, it nevertheless serves to retain bearing grease and exclude large foreign matter such as metallic chips from the bearing.

Fig. 2b—*Metal Slinger Labyrinth-Type Seal:* This seal is used where low torque, high speed, or high temperatures exist. The seal is constructed of two members rigidly fastened to the outer ring between which a steel slinger pressed on the inner ring OD is located. The slinger operates with running clearance with the other metal parts of the seal and permits high-speed operation. Since there is an all-metal seal, temperature limitations applying to rubber or felt do not exist.

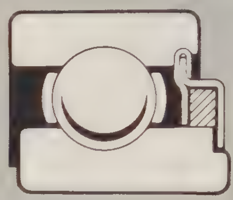


Fig. 2—*Clearance*-type seals are designed so that no contact exists between the stationary and rotating parts. Two designs of the clearance-type ball bearing seal are illustrated above.

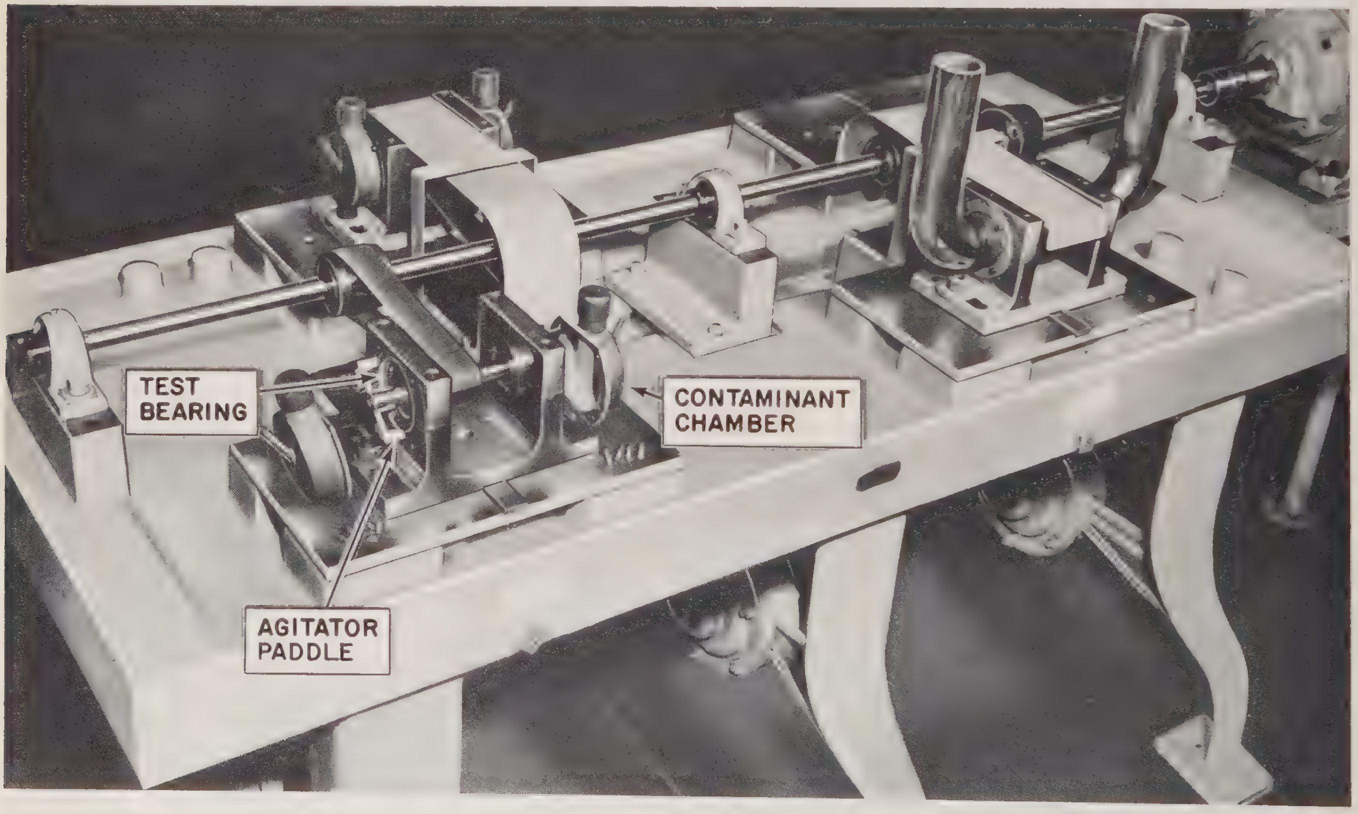


Fig. 3—A bench-test apparatus was designed at New Departure Division to evaluate integral seal bearing performance with respect to protection against contaminants and grease leakage. Bearings were mounted in the U-block supports as shown on the left. Contaminants were placed in adjacent contaminant chambers for a test run. The test device on the right illustrates the arrangement of standpipes for testing seal protection against fluids up to an 8-in. pressure head.

field experience, the relative seal efficiency of several designs, when subjected to different sealing conditions, was determined and tabulated (Table I).

Life of Sealed Bearings Depends Upon Adequate Grease Lubricant

Probably the most important factor leading to the development of sealed, lubricated-for-life ball bearings was the development and availability of an adequate grease lubricant. In the early days, the quality and consistency of available greases was such that they were entirely unsuitable when used in the small quantities required for sealed ball bearings since contaminants such as wood slivers and other foreign material often could be found in the bulk-packed grease. Further, due to oil separation, the grease consistency within a particular batch would often vary from very slushy to very dry. When an attempt was made to use greases of these extreme consistencies in a sealed bearing, either the slushy grease would quickly leak out or the extremely dry grease would be pushed aside by the rotating components of the ball bearing and be rendered less effective.

It was recognized early that a major problem would be to develop a grease which could be retained within the bearing and provide a useful, long life. Accordingly, ball bearing manufacturers pioneered in the development of a grease suitable for use in a completely sealed and lubricated-for-life bearing. When a grease with the desired characteristics was developed, the various formulas and cleaning methods were then discussed with the major oil companies for final perfection and quantity production. The millions of grease-lubricated, sealed bearings used annually attests to the success of this program.

The useful life of grease, in general, ties in with the rate at which it oxidizes. This factor is related closely to the temperature to which the grease is exposed. In general, there are two main sources of heat in a ball bearing installation: (a) that self-generated due to grease and bearing friction and (b) that resulting from an ambient condition. Although it has been found that heat developed outside the bearing affects grease life less than heat self-generated by the bearings —particularly that due to bearing loading—high temperature ambient conditions often dictate the grease life which will be obtained from a sealed ball bearing. Considerable data have been accumulated on various ball bearing lubricants, with particular attention being given to a high melting point sodium soap grease which is standard with a number of ball bearing manufacturers.

The relationship between bearing grease life and bearing outer race temperature can be shown on a semilogarithmic plot (Fig. 4). This information indicates, for instance, that at a temperature of 120° F, a life of some 21,000 hours may be expected; however, at 250° F, the expected life drops to approximately 2,500 hours. This relationship illustrates a chemical law affecting all greases, indicating in this particular case that for each 40° F temperature rise the bearing grease life is decreased by approximately 50 per cent.

Although the temperature to which ball bearing grease is exposed is the primary concern when forecasting grease life, other factors—such as speed, load and special ambient conditions—cannot be ignored. For example, test results show that the indicated life based on temperature alone (Fig. 4) must be modified by a speed factor (Fig. 5). Thus, at 120° F,

RELATIVE SEAL EFFICIENCY

SEAL TYPE	RELATIVE FRICTION CHARACTERISTICS	RELATIVE SEAL EFFICIENCY		
		GREASE RETENTION	WET CONTAMINANT EXCLUSION	DRY CONTAMINANT EXCLUSION
Felt Rubbing-Type Seal (Fig. 1a)	Moderate	Good	Fair	Fair
Felt Rubbing Labyrinth Seal (Fig. 1b)	High	Excellent	Fair	Good
Double Felt Rubbing Seal (Fig. 1c)	High	Excellent	Fair	Excellent
Removable Notch Rubbing-Type Seal (Fig. 1d)	Moderate	Excellent	Excellent	Excellent
Fixed Notch Rubbing-Type Seal (Fig. 1e)	Moderate	Good	Fair	Good
Spring Loaded Lip-Type Seal (Fig. 1f)	High	Excellent	Excellent	Excellent
Simple Plate Shield (Fig. 2a)	Nil	Fair	Poor	Poor
Metal Slinger Labyrinth-Type Seal (Fig. 2b)	Nil	Good	Fair	Fair

Table I—The characteristics of the various types of contact and clearance type sealed bearings shown in Figs. 1 and 2 are shown in this table. Seals were evaluated as a result of New Departure Division laboratory tests and field experience.

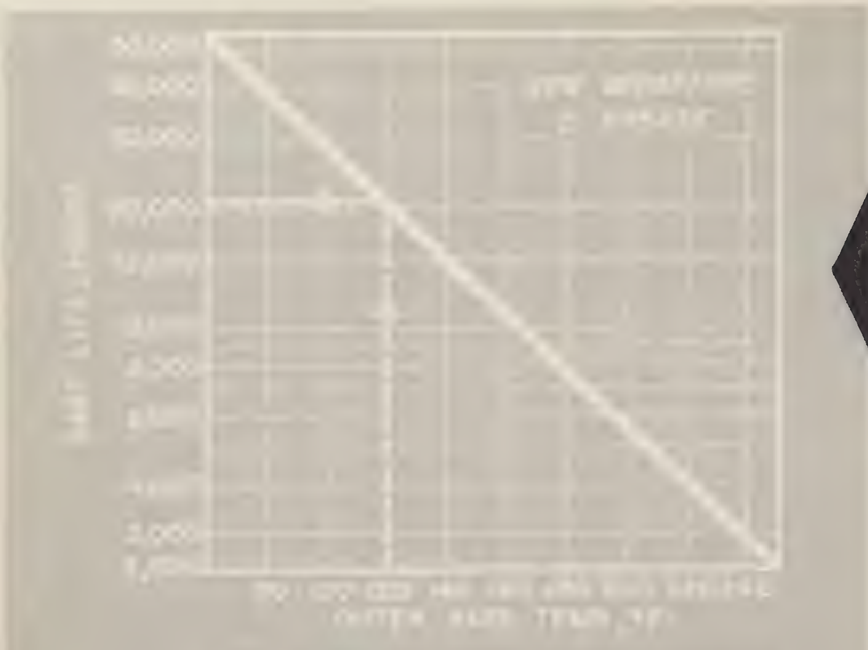


Fig. 4—The relationship between bearing grease life and bearing outer race temperature can be shown on a semi-logarithmic plot. This is a chart for New Departure C Grease in a single row, radial, non-loading groove bearing operating at 3,000 rpm, with a negligible load and ¼ full standard pack of grease. At a temperature of 120° F, for example, a life of 21,000 hours may be expected; however, the expected life is reduced to about 2,500 hours when the temperature increases to 250° F.

Fig. 5—Besides temperature, bearing speed also affects the life of bearing grease. From this chart, the designer can determine the effect of speed on grease life. Using the same example as in Fig. 4, if the speed were increased from 3,000 rpm to 8,000 rpm, the indicated grease life would be modified by a speed factor of 0.75.

Fig. 6—The distribution of bearing failures also must be considered when indicating bearing grease life. This distribution follows the typical probability curve shown above. For example, when 50 per cent of average bearing life is reached on a group of sealed bearings, approximately 90 per cent will remain unfailed.

the indicated life of 21,000 hours would be modified by a speed factor of 0.75 if a speed of approximately 8,000 rpm were involved. Likewise, the resulting life forecast often must be modified when considering bearing load and bearing size. Since these factors often introduce complex relationships, it is fortunate from the standpoint of a grease life forecast that the majority of sealed bearings operate with comparatively moderate loads.

Another factor which must be considered when discussing bearing grease life predictions is the distribution of bearing failures. This distribution follows a typical probability curve (Fig. 6). For example, when 50 per cent of average bearing life is reached on a group of sealed bearings, approximately 90 per cent of the bearings will remain unfailed. Likewise, when 100 per cent of average expected life is reached, 46 per cent of the bearings will remain unfailed. This distribution curve, of course, differs somewhat between various greases and operating conditions.

The probability curve also has been found useful in determining an optimum interval between relubrication. In this connection, it is generally recommended that a greased bearing be relubricated after a period representing 40 per cent of the expected grease life, although this procedure applies principally to removable or relube seal bearings.

Recently, considerable attention has been given to the development of a suitable and long life high-temperature grease. The standards by which the suitability of these greases are judged is often the ability not only to exhibit long bearing life at high temperatures, but also the ability to remain workable at temperatures considerably below 0° F. Although some success has been experienced in meeting each of these extreme conditions separately, the problem of developing a grease for both extremes is only partially solved at this writing.

Design Factors of the Final Device Affect Sealed Bearing Use

The determination of whether a sealed ball bearing can be successfully incorporated into a machine or device is often established on a review of the objectives of the design. If simplicity is required, even at the cost of appearance, there is good reason to believe that sealed bearings should be considered. For example, open bearings mounted in cast iron equipment, with its smoothly formed contours and neatly arranged cover plates, may present a pleasant picture to many designers when compared to steel plate construction with its bold design lines. The latter, however, housing a bearing incorporating integral seals, often represents a much more simplified and low-cost design.

Another deciding factor might be the ultimate life which is desired from the device or machine. For example, at elevated temperatures only a few thousand, or even a few hundred, hours of life can be expected from a grease-lubricated sealed bearing. If a much longer life is required, an argument against the sealed bearing would be established.

In still other designs, the availability of splash oil due to the presence of gearing or crank shafts often makes it expedient to use oil-lubricated open bearings. Conversely, if many bearings are involved and their position is scattered throughout a machine it would be very difficult and costly to consider any other arrangement than sealed ball bearings. Still other machinery, such as food and textile equipment, is heavily dependent upon the use of sealed ball bearings since even small quantities of oil dripped out of a bearing or oiling system could result in considerable product spoilage.

Although there are many exceptions, it appears that heavy, expensive machinery representing capital investment generally makes use of either oil-lubricated open bearings or single seal bearings which can be relubricated, while consumer products, especially of the expendable type, show a strong tendency to be equipped with lubricated-for-life sealed bearings.

Simplification of design, which is possible when using integral seal ball bearings, often can result in considerable savings to the equipment builder. For example, the expensive oiling systems and their accompanying problems can be entirely eliminated. When comparing integral seal bearing mounting features with other grease-lubricated designs, it can be noted that a great many components can be eliminated, such as end caps, grease chambers, and grease fittings.

Examples Illustrate Sealed Bearing Applications

Integral sealed bearings find their way into a multitude of different applications as indicated by various different seal designs, each of which has been engineered to best meet a particular service requirement. Illustrations of several typical machine designs demonstrate not only the use made of the various types of sealed bearings but also the simplicity of design made possible by using sealed bearing units (Fig. 7).

For example, in the case of a bearing housing for a textile loom mechanism, the machining required is limited to boring the housing straight through and facing one end. Integral seal bearings are used at both ends of the mounting which eliminates the need for separate seals or locating shoulders (Fig. 7a). Similarly, a double seal bearing used in a cut-off machine mounting has ample lubricant for the life of the unit and requires only a straight-bored hole for mounting (Fig. 7b). A conveyor roller mounting having a hexagonal shaft uses a bearing with a hexagonal inner ring hole and double felt labyrinth-type seals (Fig. 7c). This design affords the needed protection and results in a product which is economical to manufacture and is easy to set up and maintain.

Another example is one type of integral seal bearing application in an automotive rear wheel mounting (Fig. 7d). In this case, the bearing has one spring loaded, lip-type seal on the outside face of the bearing. This arrangement allows the bearing to be lubricated by oil from inside the differential housing while the seal provides protection from outside contaminants and eliminates the need for a supplementary seal.

These illustrations show that in each instance where integrally sealed bearings have been employed a definite saving to the equipment builder has been made possible. Further, and still more important, the user of the equipment saves on maintenance expenditures. The importance of this last factor has led to the publicizing of the term "lubricated-for-life," which most equipment builders readily employ with reference to the product whenever the occasion permits.

Maintenance Often Determines Selection of Open or Sealed Types

As stated above, an important purpose satisfied by integral sealed bearings is the decrease or entire elimination of the maintenance required for a machine or device. The exact importance of this feature can be evaluated if a comparison is made with the maintenance activities common to standard open-type bearings.

The first consideration is that of avoid-

Fig. 7—A selection of industrial applications of integral seal ball bearings illustrates some of the important design features which the use of these bearings make possible.

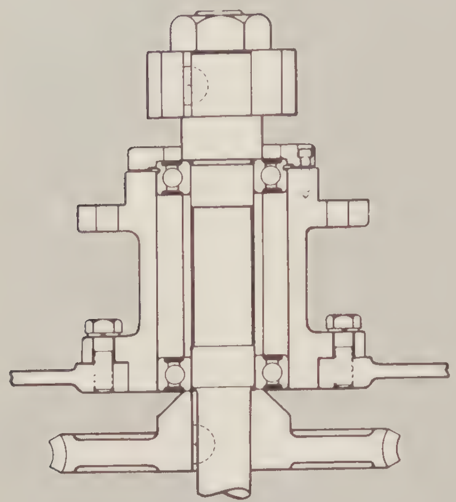


Fig. 7a—This vertically mounted textile loom mechanism utilizes two single row, sealed ball bearings. Machining is simplified because the housing can be bored straight through and faced on one end. No separate seals or locating shoulders are needed, since a snap-ring type integral seal bearing is used. Since, in general, single row sealed bearings have a thrust capacity equal to their radial capacity rating, no supplementary devices are required to accommodate the thrust loads encountered in this application.

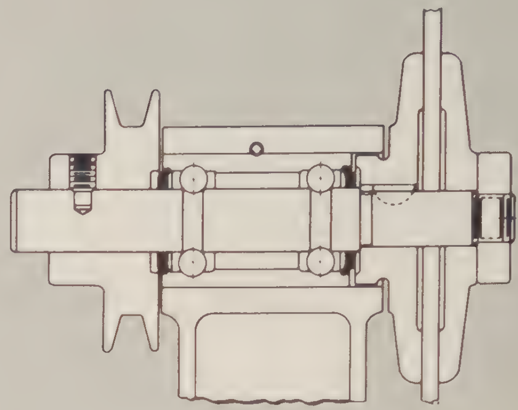


Fig. 7b—This is a cutoff machine mounting which uses a double sealed bearing of a type developed for automotive fan and water pump applications. This bearing is fitted with permanent, close fitting seals at both ends, thus eliminating any need for lubricating fixtures. An ample supply of lubricant is provided within the bearing to last for the life of the unit. As in the machine shown in Fig. 7a, the bearing housing requires simply a straight bored hole. A snap ring, engaging the bearing outer ring, positions and locates the bearing with respect to the housing. As a result of using this particular sealed bearing, there are several advantages in the design, such as (a) a smaller number of parts, (b) easier, more accurate machining, and (c) simpler, quicker assembly with service requirements practically eliminated.

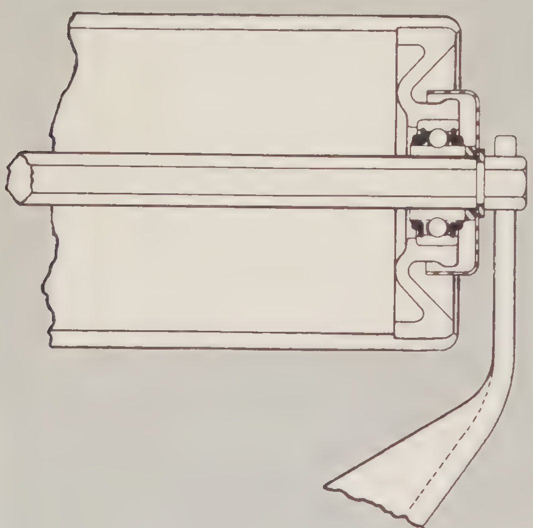


Fig. 7c—Another special purpose integral seal bearing is represented by this application in a conveyor roller. The bearing has a hexagonal inner ring hole which fits shafting of the same form. Double felt labyrinth seals are provided and this, together with the supplementary deflector shown, affords maximum protection. The simplicity of the design not only enables the conveyor manufacturer to produce the roller economically but also results in units which are extremely easy for the operator to set up and maintain in the field.

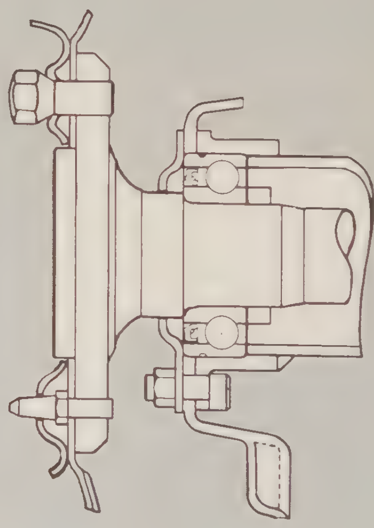


Fig. 7d—An automotive rear wheel mounting is another application which demonstrates some of the advantages of the integral seal bearing. In the installation shown above, a single, spring-loaded, lip-type seal is used on the outside face of the bearing. This permits the bearing to be lubricated from the opposite side by the oil employed in the differential housing. This same seal provides sufficient protection from outside contaminants to eliminate the need of any other supplementary seal. Automotive designs vary and this rear-wheel application often utilizes ball bearings which are double sealed and prelubricated with a suitable grease.

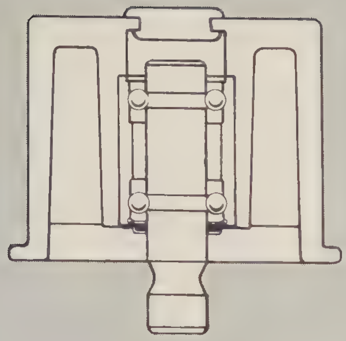


Fig. 7e—A simple method for mounting an idler pulley is illustrated by this application of a double row bearing using a special purpose, prelubricated single seal. Lubricant is contained within this bearing, but provision also is made for adding a small quantity of oil at suitable intervals to rehabilitate the original grease. This feature usually is necessary to meet the extremely long life requirements common to such fields as the textile industry from which this example is taken. The simplicity of design and small number of component parts result in considerable saving to the builder of this type of equipment.

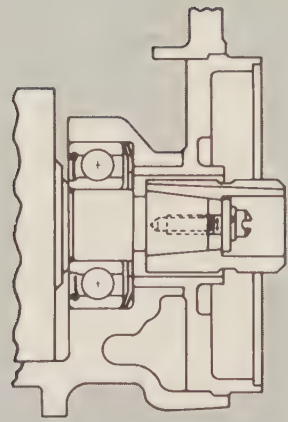


Fig. 7f—Shown here is an aircraft magneto application of the integral seal bearing. This bearing is grease lubricated and has a single seal of the *clearance*, or non-contacting type. The features shown are typical and can be applied to many designs using either single or double seals.

ing dirt or other foreign material before it gains entrance into the bearing to cause wear, destroy accuracy, or otherwise shorten life. Ball bearing manufacturers observe the strictest care at all times to keep foreign material out of bearings. However, after delivery to the user the success of keeping the bearings in this desirable condition depends upon the care taken in storage, handling, and the methods used during their installation.

In a double seal bearing the seals themselves assure ample protection to the internal surfaces of the bearing during storage, even if the bearing box or wrapping becomes damaged. It also is highly important to prevent dirt, which accumulates on assembly benches or mounting tools, from entering open-type bearings during the mounting operation. Obviously, necessary precautions against these conditions play a decidedly minor roll when double seal bearings are involved.

Another phase of bearing maintenance involves lubricating the bearings after the machine is set up for operation. In the case of sealed lubricated-for-life bearings further attention to this factor may be ignored. With open bearings much attention must be given to such tasks as changing and replenishing the oil supply,cleaning and filtering the oil during circulation, cleaning out oil sumps, and maintaining the pumping system.

Considering the various activities involved in the preventive, operating, and up-keep phases of maintenance for open bearings, it is no wonder that so many designs have been converted to make use of bearings which are sealed and lubricated for life. Unfortunately, each and every design does not lend itself to this simplification. A bearing often must be protected against especially severe contamination conditions, and this can be best handled by a separate commercial seal. In addition, other operating factors often dictate the use of circulating oil, in which case either open bearings or appropriate single seal bearings are employed. In recent years the latter case has undergone considerable development brought about by improved and efficient synthetic rubber-type seals made of materials impervious to damage from oil exposure.

Future Trends of Interest to Designers

From the seal builders' standpoint, the trend is toward a single, all-purpose seal. While recent developments in design and material accomplish this objective to a limited degree, the ideal seal is envisioned as being effective for all types of contaminants and under all operating conditions. Furthermore, the seal should last at least as long as the bearing and be inexpensive to produce.

Users of sealed bearings indicate a definite trend toward higher temperatures. For this condition, such new synthetic materials as teflon and silicones are being considered. Their high cost, rigidity, and low resistance to certain lubricants, however, still are drawbacks at the present time.

The need for low torque is another seal characteristic which is receiving considerable attention. The high operating speeds encountered in many recent applications require a seal that will generate a minimum of heat, since high operating temperatures deteriorate the seal material and also serve to reduce the useful life of the lubricant.

To improve the efficiency of the integral seal, it has been found necessary to improve its precision. This has been accomplished by a relatively low cost molded sealing element made from synthetic rubber. Designs using the molded construction satisfy other requirements recently brought into prominence, namely: (a) the seals can be removed and replaced, (b) bearing ring distortion is kept to a minimum, and (c) seals are not loosened by exposure to severe vibration.

Acknowledgement

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The Development of the General Motors Air Spring

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General Motors
Engineering Staff

A piston and air chamber device gave satisfactory spring rate and mechanical characteristics

Ride improvement in automobiles has been sought continually by engineers through the use of different types of steel springs such as single leaf, torsion bars, coils, and other devices. One limitation of steel springs that has been most bothersome to designers has been that curb standing height and full-load ride clearance vary when the vehicle load varies, as in the case of full baggage and full passenger loads. These conditions do not allow best riding qualities. The present automotive design trend toward lower car heights and lower car floors has intensified the problem. GM engineers, believing that a fundamental change in springs was necessary, directed their attentions to the long-known principle of the *air spring*. As a result, the GM Engineering Staff (located at the GM Technical Center) developed an air spring device which satisfied the following criteria: reasonable cost and size, long life, adaptability to any type of suspension, adaptability to any desired length of stroke, and correct spring rate characteristics. This design was then further developed and utilized by the Cadillac Motor Car Division on the 1957 Cadillac Eldorado Brougham.

THE riding characteristics of automobiles have undergone gradual improvement over the years as a result of the continuous efforts of engineers to develop new suspension systems and new components. A system for supporting and controlling the vehicle, a suitable spring, and a damping, or shock absorbing device, have been the basic elements of the automotive suspension. These elements have benefited from improvements in design and in materials so that the passenger of the modern car enjoys a ride which minimizes objectionable motions such as bounce, sway, roll, and vibration. Some form of a steel spring, such as a leaf spring, coil spring, or torsion bar has been a satisfactory device and has undergone many minor improvements.

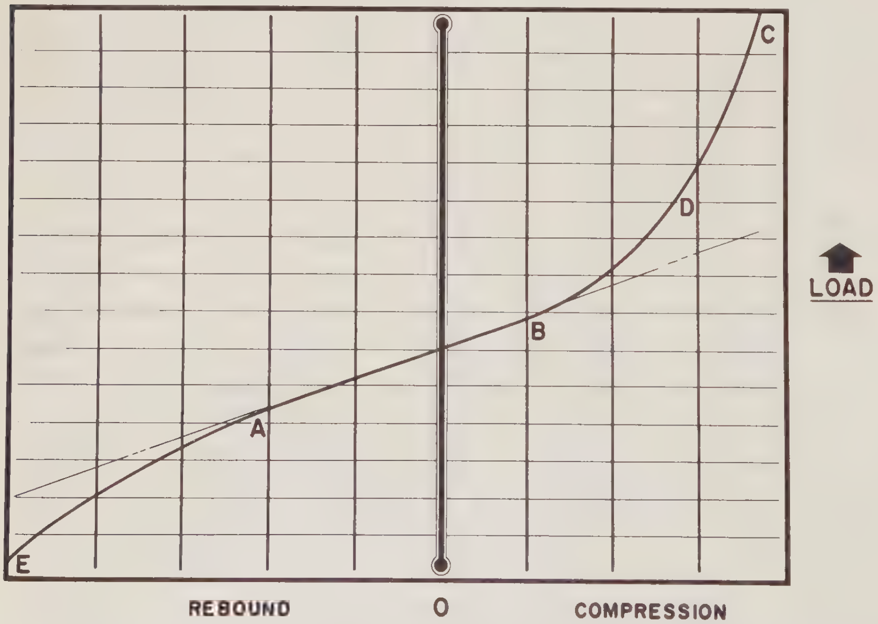


Fig. 1—The ideal shape of the load-deflection, or spring rate, curve is shown above. For some distance to either side of point *O*, the normal position, the curve is a straight line indicating that the spring neither gains nor loses rate. When coupled with a standing height adjustment, this characteristic assures a consistently balanced ride. A high end load, at *C*, reduces any sensation of "crash through" when riding over ruts or holes. A gentle transition, B-D-C reduces abrupt swells in the ride. It also is desirable to seek a considerable reduction in load on rebound, at *E*. This condition reduces lift when braking and accelerating, and prevents excessive input to the frame and suspension members. These spring characteristics were considered desirable in the development of a new air spring device.

Engineers have recognized, however, that steel springs have certain limitations[1]. Of great concern has been the limitation that steel springs are not readily adaptable to increased static deflections without affecting the relationship between curb standing height and full-load ride clearance. This condition becomes an even more important problem today because of the automotive industry's trend toward lower car roof heights and lower floor heights. The search has been for a solution which would permit maintaining a soft ride without undesirable compromise.

Some years ago, the General Motors Engineering Staff approached the problem from the standpoint that a fundamental change was necessary. A study was made of air as a suspension medium, which in principle was not a new idea even though it had not been adopted on American passenger cars for various reasons. The result was the type of air spring which now has been adopted for use on the 1957 Cadillac Eldorado Brougham.

Ideal Load-Deflection Curve Guides Air Spring Design

The GM developmental program began with establishing the following design criteria for the air spring:

- The spring must be of reasonable cost and size
- It must have long life
- It must be adaptable to any type of suspension
- Its fundamental concept should permit any desired length of stroke to be obtained without altering other design features
- It should have the desired spring rate characteristics.

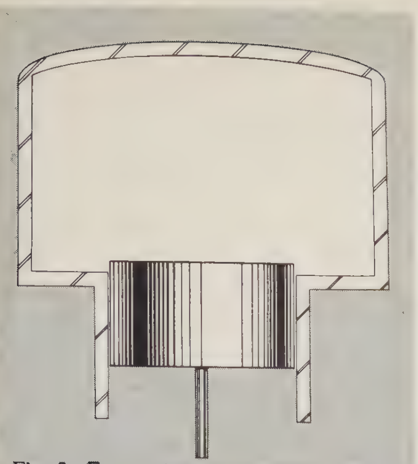


Fig. 2—For mechanical simplicity, and to achieve the desired spring rate, attention was given to some form of the fundamental piston and cylinder-type air spring. The load carrying capacity is determined by multiplying the piston area by the gage pressure.

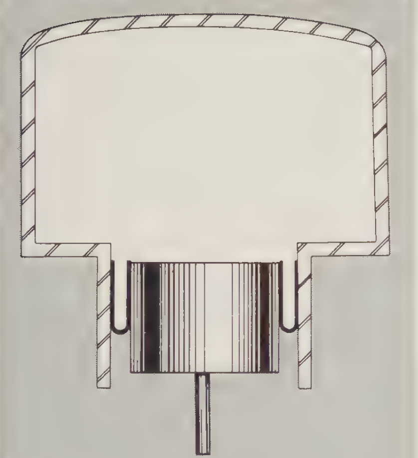


Fig. 3—In a practical automotive installation, an adequate air seal could not be obtained in the fundamental type shown in Fig. 2 without excessive cost penalty. A wide-clearance design having a rolling seal was then proposed. This type would eliminate any problems of sealing and of manufacturing tolerances.

The spring rate characteristics were studied on a load-deflection curve representing an ideal shape (Fig. 1). It was considered that the spring should neither gain nor lose rate for a considerable distance on either side of the normal position. This characteristic, when coupled with some form of standing height adjustment, permits a constantly balanced ride over major road irregularities. The spring also should have a high end-rate to reduce any riding sensation of "crash through" when passing over ruts or holes. The transition between the low center-rate and the high end-rate should be gradual, to avoid upsetting the ride when passing over abrupt swells.

Previously, the air spring used successfully in the suspension system of the General Motors Coach[2] was studied for possible application to passenger cars. This spring is a double convolution bellows-type. Investigation showed that to obtain the necessary spring rate for passenger cars, an additional air reservoir of undesirable size would be required. For long-stroke, direct-acting rear axle applications, a much longer bellows design would be needed. By increasing the number of convolutions, however, the instability of the spring would be increased. Even assuming that such mechanical difficulties could be overcome in some manner, the resulting load-deflection curve would not have the shape as determined in the original criteria for the passenger car spring. Thus, it was concluded that a scaled down version of the successful bus air-spring design would present more undesirable characteristics than could be tolerated.

Piston and Air Chamber Best Design for Passenger Car

For the passenger car application, therefore, engineers could rule out using some form of the existing GM device. Attention was then directed to the fundamental form of an air spring, consisting of a piston compressing air in a chamber (Fig. 2).

The load carrying capacity of this spring is a matter of simple mathematics: area of piston multiplied by gage pressure equals load. Applying the tight-fitting piston and chamber principle immediately presented certain practical problems. It would be difficult to obtain a 100 per cent seal against air leakage without paying an excessive penalty, both in friction and in dollar cost for precision machining.

The next design approach was to increase the piston clearance and install a rolling seal (Fig. 3). At one stride, two problems were eliminated: (a) minor size variations of the piston and cylinder were inconsequential, and (b) friction was

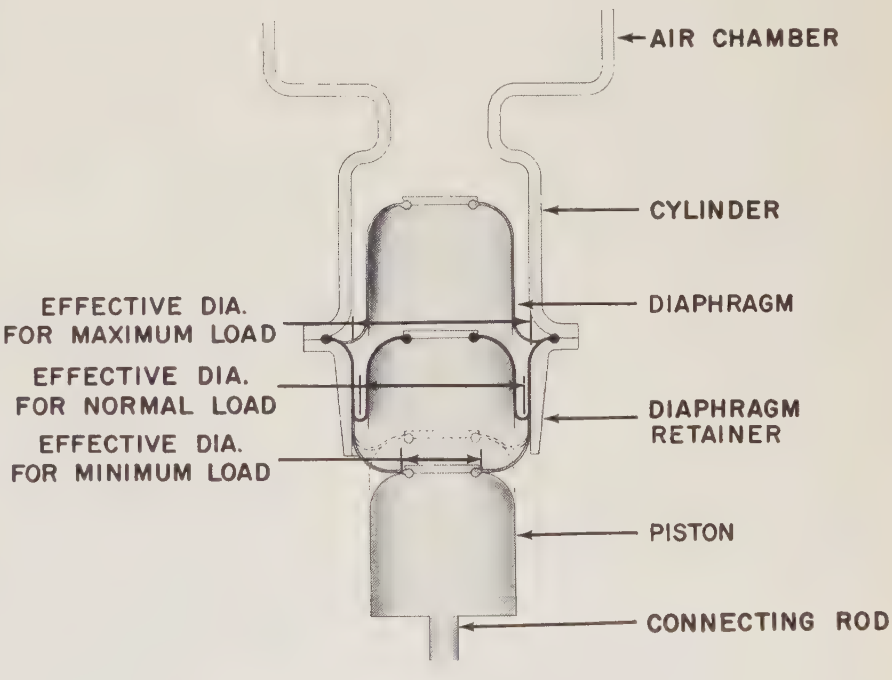


Fig. 4—In the theoretical design of the air spring, it was recognized that a piston with a variable area would be needed to produce the gentle buildup and the high end load as determined from the ideal spring rate curve of Fig. 1. A variable area arrangement can be achieved by the above design. A constant effective diameter is maintained for some distance above and below normal position. An increased diameter (hence increased area) is obtained in the upper position. A reduced diameter is obtained in the lower position when the diaphragm moves away from the piston. In all positions, the length of a horizontal line, tangent to the meniscus, determines the effective diameter. The percentage increase in effective area in the full compression position is less than the decrease in full rebound. This is necessary due to the fact that the air pressure increase, as the piston enters the chamber, tends to assist in load buildup. In rebound, the air pressure drop requires a more rapid reduction of effective piston area. The above illustration also shows one method of attaching the diaphragm to the piston and chamber. A wire bead ring, encased in rubber, seals with a clamp plate to the piston and also is clamped between the outer rims of the air chamber and diaphragm retainer.

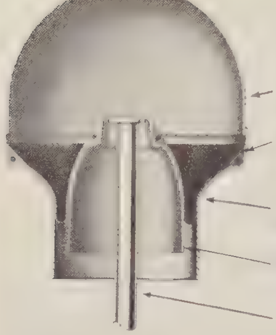


Fig. 5—Another method of attaching the rubber diaphragm is shown in this final version of the design of an individual air spring. Steel bead rings are surrounded by wedge-shaped rubber lips which fit into properly shaped portions of the piston and retainer. The internal air pressure on the rubber lips forms the seal.

eliminated. Furthermore, the virtues of the original piston concept were retained.

The diaphragm presented no unusual problem of manufacture. A backlog of experience in tire making by the supplier company solved most of the difficulties connected with making a rubber impregnated fabric which could withstand flexing to a practical degree. No air springs were ever made, of course, with the diaphragm in the initial shape proposed.

While the simple, rolling seal design represented the basic concept of the straight piston and cylinder it did not satisfy completely the conditions laid down in the ideal spring load-deflection curve. The long low rate could be obtained, but gentle buildup and the high end load were missing. A variable area piston was required. This was obtained by increasing the outside diameter of the diaphragm and adjusting the piston and retainer to permit a regulated adjustment of effective piston area (Fig. 4).

With this early hypothetical spring, the diaphragm was restrained between the piston and the outer skirt, or diaphragm retainer; thus, the effective diameter was fixed for some distance below and above the normal position. When the diaphragm was moved upward from its previously restrained positions an increase in effective diameter and, therefore, an increase in effective working area was obtained. Conversely, when the piston was moved to its lowest position the diaphragm lifted away from the piston and produced a greatly reduced effective diameter.

With a basic theory of design for an air spring established, the next step was to develop a method of attaching the diaphragm to the piston and air chamber. Two methods were employed, both equally acceptable. The first method used a circular wire bead ring encased in rubber which sealed with a clamp plate to the piston, and between the skirt and chamber at the outer rim (Fig. 4). This formed a very satisfactory connection, and was used for some time in the early stages of the design program.

The second method of attachment again used steel bead rings but a wedge-shaped rubber lip surrounded each ring (Fig. 5). The internal air pressure on the

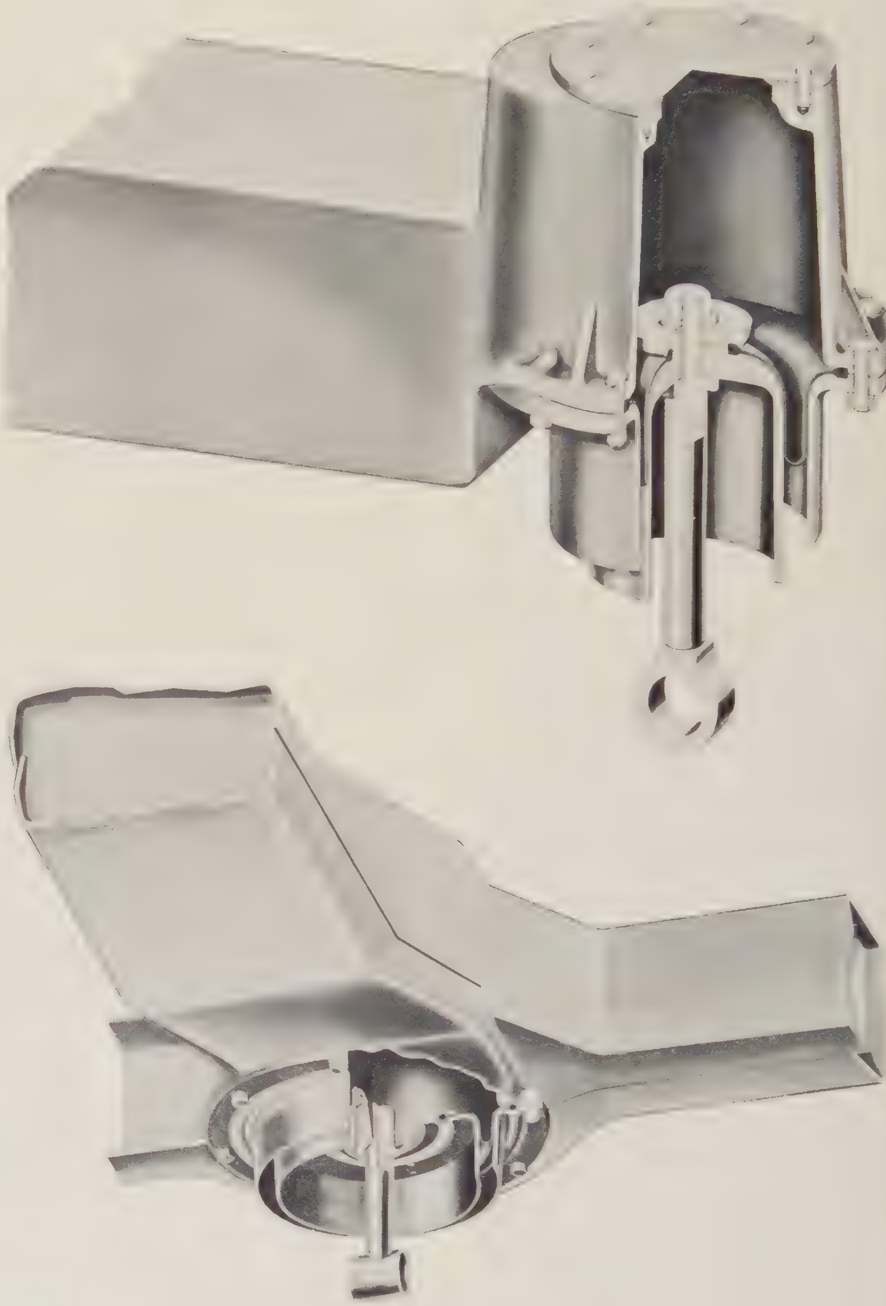


Fig. 6—As part of a vehicle testing program, which accompanied the theoretical studies, two types of air chamber arrangements were built to utilize interior space in the frame members. Shown above is an experimental spring using an aluminum frame cross member. This design resulted in certain disadvantages, such as porosity and a reduction in car frame stiffness, in spite of the use of multiple attaching screws. Another type of integral frame air chamber is shown below in a steel cross member. This type had such disadvantages as unreliable seal adhesion, leakage caused by frame motions, difficult field repair methods, and lack of interchangeability between different car models.

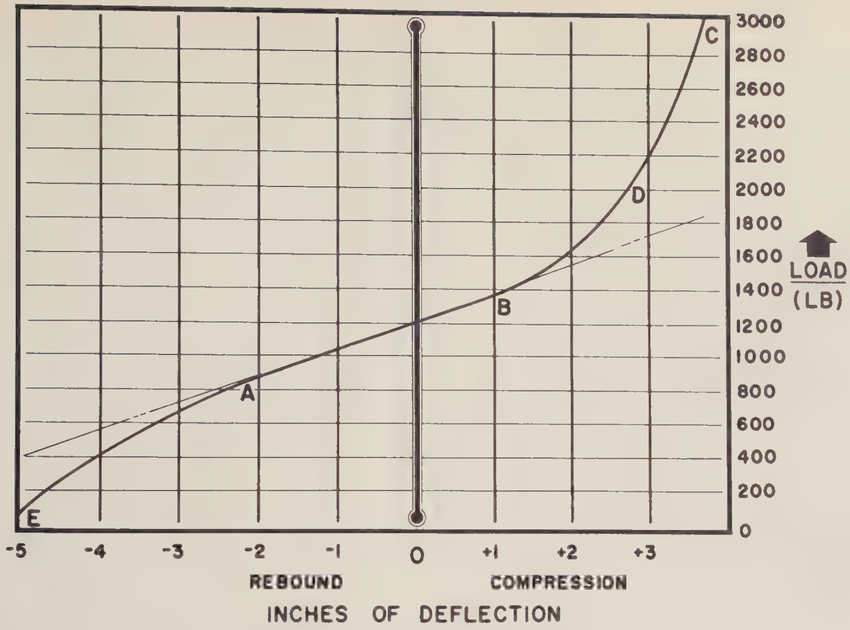


Fig. 7—Shown here is the load-deflection curve used in the development of an actual rear air spring.

rubber lips formed a seal in much the same manner as a tubeless tire to its rim. The diaphragm was installed in the retainer by temporarily deforming the normally circular outside bead into an oval shape that could pass upward through the circular opening of the retainer.

Air Chambers Tested in Experimental Cars

Concurrently with theoretical studies, road test evaluation was instituted. In this program, a number of experimental suspension components were built and tested on the vehicle. For example, consideration was given to utilizing the interior of frame cross members to obtain the necessary air chamber volume. One road car used hollow aluminum cross members, front and rear, bolted to the frame (Fig. 6—top). This idea presented seeming advantages. They were not borne out, however, in practice. In spite of a multitude of attaching screws a reduction in frame stiffness was experienced, in comparison with the conventional all-steel frame.

Drawing on this experience, a second car was tested using integral steel cross members containing the air chambers (Fig. 6—bottom). This car was sent through the standard General Motors Proving Ground 25,000-mile durability test primarily to obtain information on cold weather operation over gravel roads and in slush, ice, and snow. The chambers were built to simulate what could be expected from production tooling, with production limits, with the intention of introducing a sealer similar to that used in aircraft fuel cells. Again, practical difficulties arose. The sealant was unpredictable in its adhesion, frame motions tended to cause leaks, field repair was difficult, and probably the greatest hurdle of all—there was inflexibility of chamber size alteration between car models.

The air chamber design integral with the vehicle frame then was rejected and consideration was given to individual air chambers. The result was an air spring design consisting of a piston rod, piston, rubber diaphragm, and a diaphragm retainer and chamber housing of relatively simple shapes (Fig. 5). In the experimental version, the housing and retainer were fabricated by the metal spinning process and welded together. The following advantages were seen in this air spring design:

- Simple shapes with reduced amount of welding
- Absence of structural stresses in chambers
- Adaptability to bench testing
- Ease of installation and field service
- Interchangeability with optional steel springs.

Theory Applied to Design Spring with Proper Rate

In the theoretical design of this spring, the desired load-deflection curve was the basic consideration (Fig. 7). Knowing the load, operating pressure, and desired rate, the effective piston diameter was readily calculated. Clearance between the piston and retainer was established at a figure found from tire building experience to be satisfactory for long life.

The volume of the chamber was obtained from the following formula:

$$R = \frac{(\gamma)\ (P)\ (A^2)}{V}$$

where:

R = rate (lb per in.)
γ = adiabatic factor
P = absolute pressure (lb per sq in.)
A = effective piston area (sq in.)
V = chamber volume (cu in.).

This, of course, applied only to the derived straight section (A-B, Fig. 7) of the curves, from the basic equation:

$$R = \frac{dw}{dh} = (P - 14.7)\frac{dA}{dh} + A\frac{dP}{dh}.$$

Gamma is accepted as 1.4, the usual adiabatic factor for diatomic gases. But caution should be exercised. Under certain con-

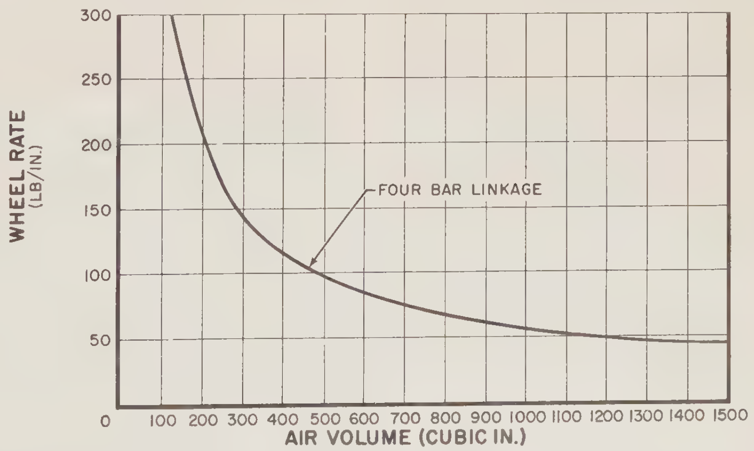
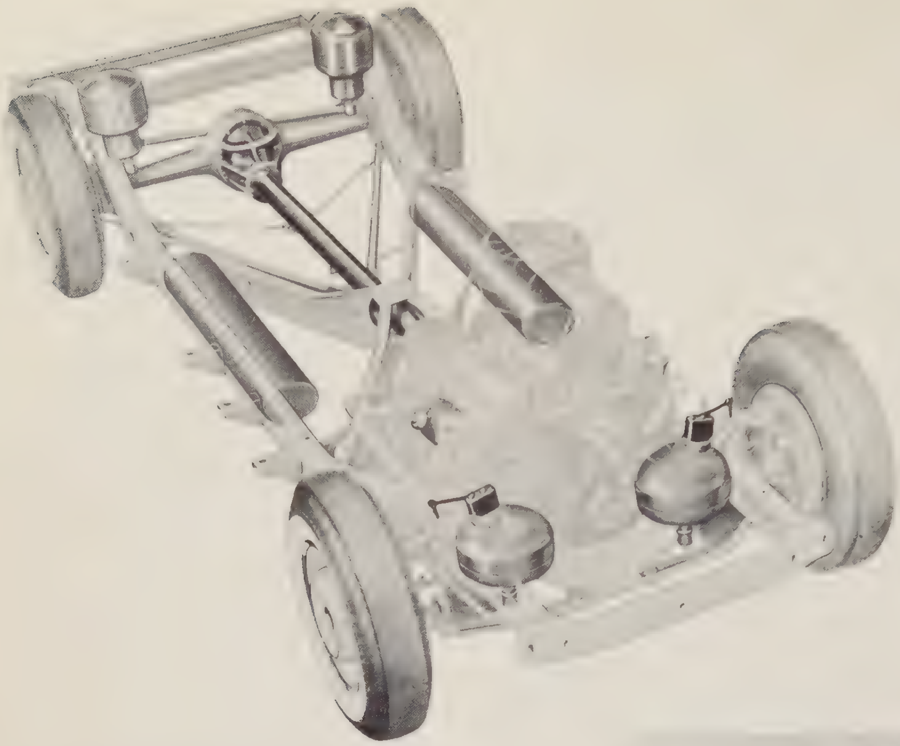


Fig. 8—A comparison of the wheel rate and the air volume required is shown for the air spring design. It demonstrates how a small reduction in wheel rate would require a relatively large increase in air volume emphasizing the need to design the air spring at a rate as close as possible to the required wheel rate.



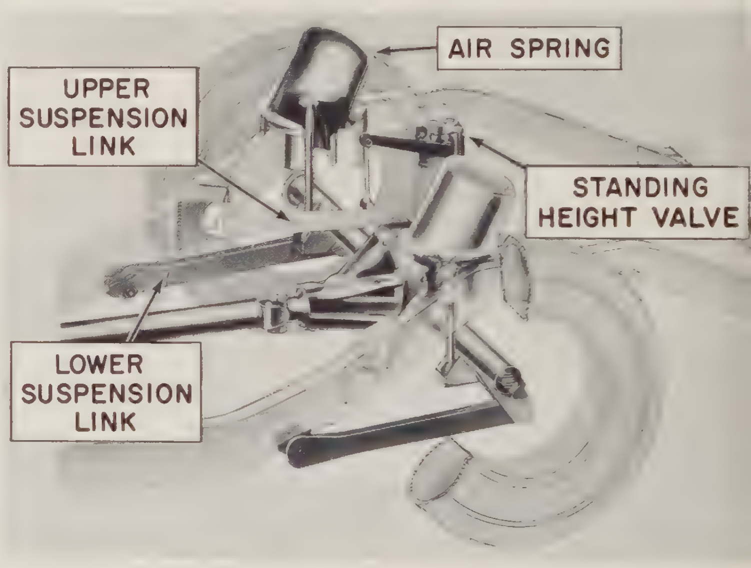


Fig. 9—The individual air springs were tested on cars with two types of rear wheel suspensions. The air spring is readily adaptable to the torque tube drive chassis (above). A four-link type rear suspension is shown with air springs at the right. The air piston rods are attached directly to the axle. No major frame alterations are necessary when using either type of rear suspension.

ditions, the air spring may function in a region approaching isothermal operation, with the attendant lower rate.

During the early development of the diaphragm spring, it was felt that the curve should show a constant rate of acceleration in the transition and high end-rate sections (B-C, Fig. 7). Later, a simpler method of calculation was used. When the straight section of low rate (A-B) was established, the desired end rate was placed on the graph. A family of curves had been developed which, from experience, were recognized as giving a desirable blend and end rate. The same procedure was followed on the unloading portion (A-E, Fig. 7) of the curve.

In the examples shown, the upper portion of the curve was calculated and found to be a fifth order equation:

$$W = 1{,}227 + 163.6\,h + 1.80\,h^5$$

where:

$$0 \leqq h < 3.66.$$

For the unloading portion of the curve the equation was:

$$W = 1{,}227 - 163.6\,h - 2.4\,h^3$$

where:

$$0 \leqq h < 4.91.$$

When the values were fed into a digital computer, the profiles necessary for the piston and retainer were established in ten minutes time. Without the computer, two mathematicians would have required two days to complete one calculation series.

A study of the change in air volume required to obtain a change in wheel rate indicated why the air spring should be designed to be in close proximity to the spring rate expected for its final form. A 250 per cent increase in volume would be required, for example, to effect a 50 per cent reduction in wheel rate (Fig. 8).

Selection of the suspension and application of the air spring is the perogative of the GM car division which decides to use the system. The Engineering Staff, however, investigated many types of rear suspensions on test cars during its developmental task.

The torque-tube drive was readily adaptable to air suspension and required no major frame alterations to accommodate two rear air springs (Fig. 9 left).

Since absence of leaf springs from a Hotchkiss suspension removes the axle fore-and-aft and lateral positioning control, a four-link system was devised, which was adaptable to the air spring with a minimum of frame changes (Fig. 9 right).

Conclusion

From this developmental program, it was concluded that the diaphragm type of air spring met satisfactorily the design criteria established. It was a device of reasonable cost and size, it was capable of long life, it was adaptable to any type of suspension, its stroke could be varied easily, and it produced the desired spring rate characteristics. The air spring was then adapted by Cadillac Motor Car Division to its suspension design for the 1957 Eldorado Brougham.

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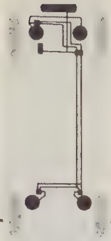
2. Fox, Harold E. and La Belle, Donald J., "Air Suspension Ride: How Engineers Gave Resilience to a Compressed Air System with Rubber Bellows," *General Motors Engineering Journal*, Vol. 1, No. 2 (September-October 1953), pp. 40-44.

Applying the Air Spring Design to the Suspension for the Cadillac Eldorado Brougham



By FRED H. COWIN
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Cadillac Motor Car Division

Air suspension systems have been used on some American trucks and buses for several years. Because of the different engineering design problems in passenger car applications, however, these air suspension systems could not be simply scaled down to become passenger car versions. Thus, General Motors continued its interest in air suspension systems for passenger cars. With the development of an air spring by the GM Engineering Staff, Cadillac Motor Car Division utilized this device in developing the air suspension system for the 1957 Cadillac Eldorado Brougham. Mounted at each of the four wheels is an air spring consisting essentially of a dome air chamber, a rubber diaphragm, and a piston. Other principal components are an air compressor and accumulator, necessary air piping, leveling valves, and solenoid controls. The new system provides the constant-height feature, uses the ride clearance more effectively, and makes possible lower ride frequencies for greater passenger comfort.

"Riding on air" is achieved with simplicity, durability, and dependability

AIR-SPRING suspension designs have been of interest to Cadillac Motor Car Division engineers for some years. The advantages of air springs, borne out by the experimental development of the GM Engineering Staff, became particularly attractive when the 1957 Cadillac Eldorado Brougham was being designed.

The general features of the air spring which were important to the Eldorado Brougham included the following:

- Constant height could be maintained regardless of the load conditions of the car. The importance of this feature is emphasized by the lowered car height of the Eldorado Brougham. It is only 55½ in. high—a drop of 3½ in. from the other 1957 sedan models and a drop of 6½ in. from the 1956 models (Fig. 1).
- The height available for ride clearance between the axle and the frame could be used more effectively. A constant ride clearance under all load conditions could be maintained, and the ride rate and balance would not change from minimum to maximum load.
- Lower ride rates would be possible because the constant-height feature would give a constant-ride travel. This would allow the designers greater latitude in selecting the best characteristics for both ride and handling.

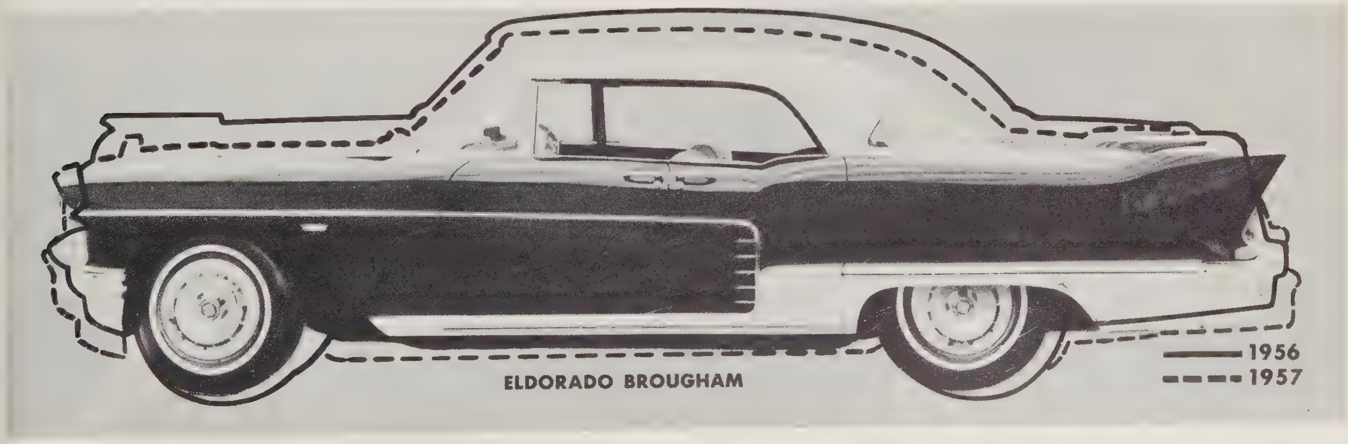
Design Goals Established

Knowing these promising advantages of the air spring, and supported by further experimental testing on modified Cadillac production cars, the following design parameters for the new Eldorado Brougham springing were established:

- *The ride rates would be approximately 55 cycles per minute, or about 10 per cent lower than those existing with Cadillac steel-spring suspensions.* These rates were considered to be a good compromise for handling and ride since lower ride rates might introduce problems in handling.
- *The diaphragm-type of air spring would be used rather than the bellows-type as used on buses.*
- *Air dome pressure at normal passenger load would be approximately 75 psi.* This would permit a piston and air dome of reasonable size that could be accommodated conveniently in the chassis.

*For Mr. Milliken's biography and photograph, please see p. 63 of the April-May-June 1957 GENERAL MOTORS ENGINEERING JOURNAL. This is the second paper contributed by Mr. Milliken, staff engineer in the Engineering Department of Cadillac Motor Car Division.

Fig. 1—The design of the 1957 Cadillac Eldorado Brougham (below) called for a lowered car height of 55½ in. This new height is compared with outlines of comparable car models in the standard 1957 Cadillac series (dotted outline) and in the 1956 Cadillac models (solid outline). The lowered height was one of the reasons which led designers to the use of the air suspension system. With the lower height, the ride clearance between wheels and chassis would be lower; thus, air springs could utilize the clearance more efficiently and maintain a constant height at all conditions of car loading.



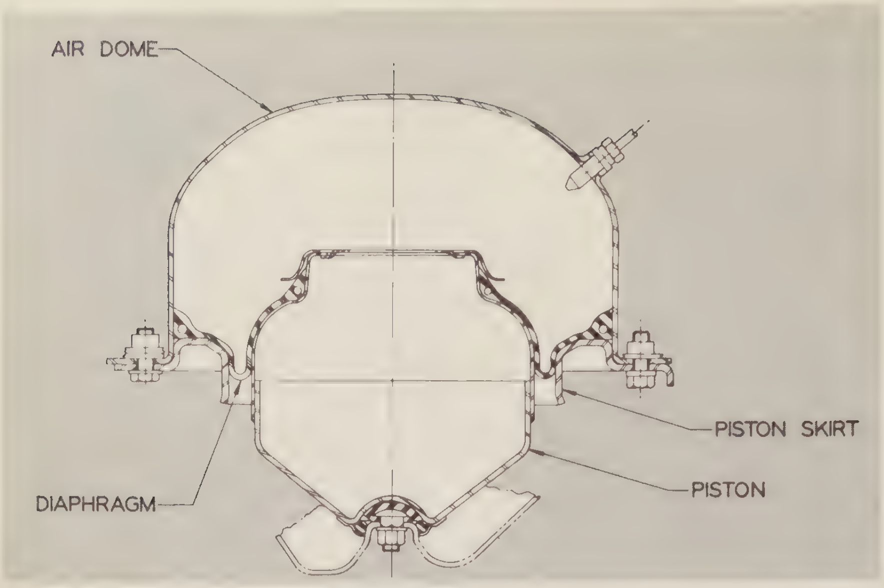


Fig. 2—The final design of the front air spring, illustrated here in the normal height position, utilizes a hollow piston having a 3-in. diameter hole in the top which provides part of the required air volume of the spring. This allowed a smaller size spring which could fit into the frame cross member. The piston, mechanically retained to the diaphragm by a metal clip, is attached to the lower control arm of the front suspension by a bolt. The bolt head is encased in a phenolic resin cover having a spherical shape to match the seat in the bottom of the piston. A rubber ring is held against the bottom-seat depression by a metal retainer which prevents the piston from pulling off of the lower control arm at extreme rebound height. The rubber ring also is an efficient seal for lubricant at the seat. The air line connection and valve is shown in the top of the dome.

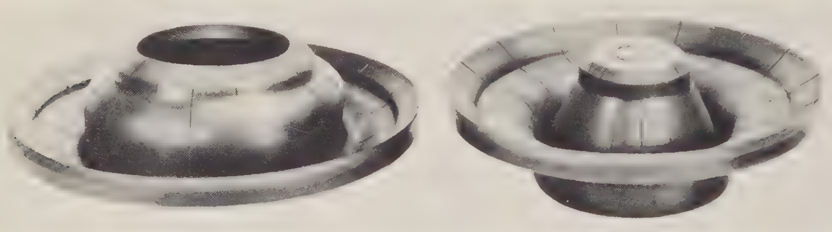


Fig. 3—The rubber diaphragms used in the front (left) and rear (right) air springs are similar but not interchangeable. The diaphragm is of two-ply construction with solid wire beads, which were found to be superior to braided wire for a tight seal. Natural rubber is used because of its good flex life. Developmental work in conjunction with the diaphragm supplier resulted in a wall thickness of about 1/8 in. for the section between the beads. This is thin enough to assure a low strain in the outer skin of the diaphragm. There is, however, enough rubber to assure ample coverage of cords during the molding process. On the rear diaphragm (right), a steel washer is vulcanized into the solid center. A stud on the under side of the washer screws into the tapped head of the piston. This provides a mechanical connection between the diaphragm and the piston.

- *The front suspension would be a modification of the basic Cadillac parallel-link design, with air springs replacing the usual coil springs.* This would permit considerable interchangeability of frame and suspension parts with the standard line of cars, which would continue with coil-spring front suspension.
- *The rear suspension would be of the four-link type with a rigid axle.* This would be entirely new and different from the leaf-spring and Hotchkiss-drive design used on the standard line of Cadillac cars. The four-link design would permit independent control of roll center, lateral stiffness, and acceleration squat. The roll center could be raised and established definitely to assure proper car handling with the proposed lower spring rates. Lateral stiffness could be increased to reduce rear-end sway and steer. Understeer could be set and depended on, as the suspension height would not change with the load. The swing arm for the rear axle could easily be set for the best performance between rear-end rising or lowering on acceleration, a latitude that was not possible with the leaf-spring design. This type of suspension would give better control of rear axle pinion wind-up under both braking and acceleration torque.
- *The air system would be of the open-type.* Inlet air for the compressor would be drawn from the atmosphere. Vented air from the springs would be discharged to the atmosphere. This system differed from the *closed* type used in the GM Engineering Staff design, which returned the used air to a low-pressure tank from which it was drawn by an air compressor for re-use in the air spring. An advantage of the proposed open system would be the elimination of a low-pressure tank and associated plumbing.

With these major parameters established, design work was started to adapt the basic design to the chassis require-

Fig. 4 (right)—A test apparatus was built to evaluate the sealing qualities of rubber against various types of steel surfaces. Rubber suction cups, 3 in. in diameter, were attached to each test specimen. Springs, loaded to 4 lb, pulled the suction cups from the surfaces. The time required to break the seal, indicated by timers, was a measure of the air-tightness of the seals.

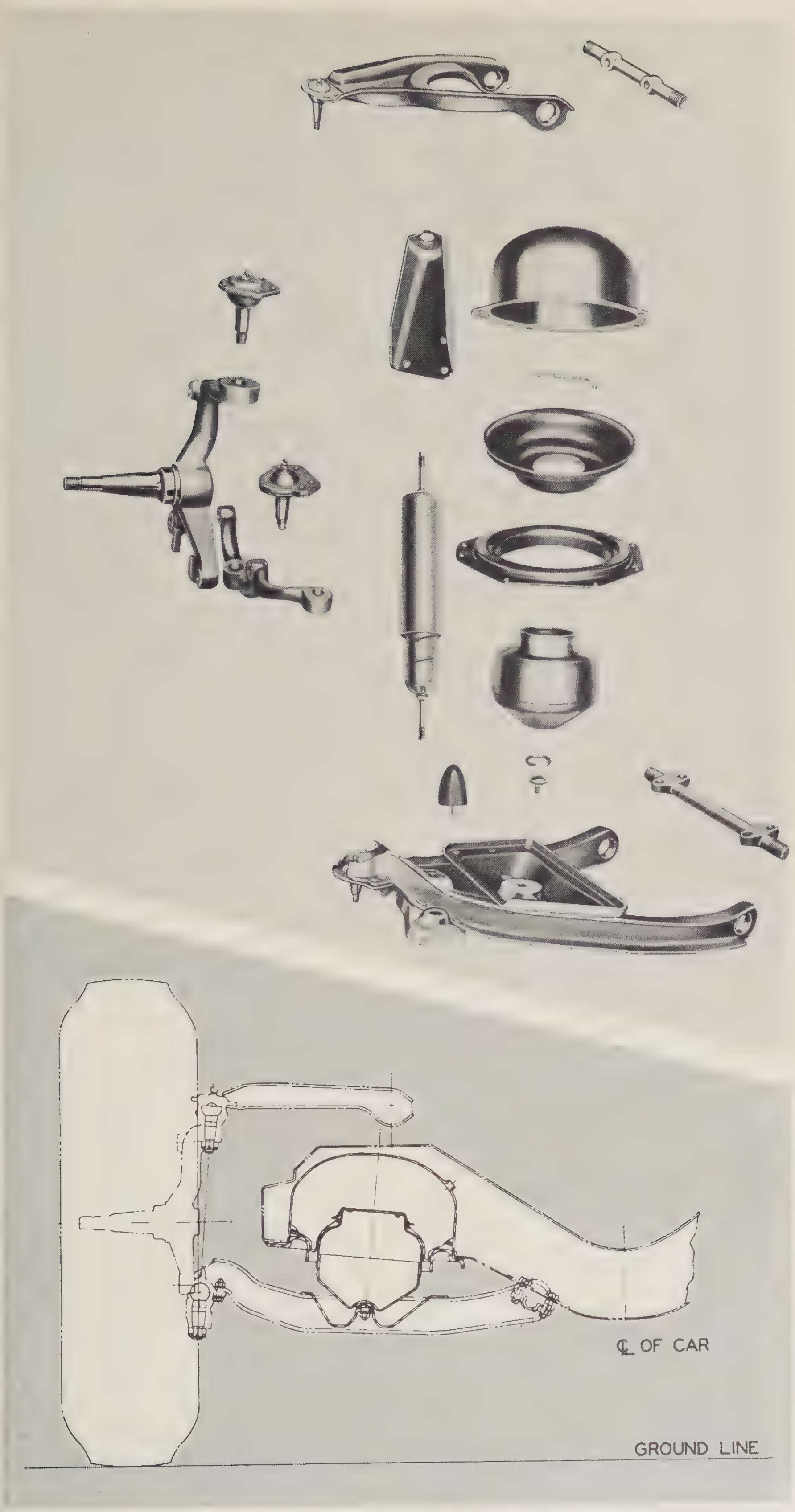


Fig. 5—The exploded view shows the relation of the air spring components to other parts of the front suspension. The shock absorber is attached to a bracket on the lower control arm and to an upper bracket bolted to the frame side bar. The diagram below indicates the position of the assembled spring. The air dome is bolted to the under side of the frame cross member.

ments of the Brougham body.

Front Air Spring Adapted to Existing Suspension

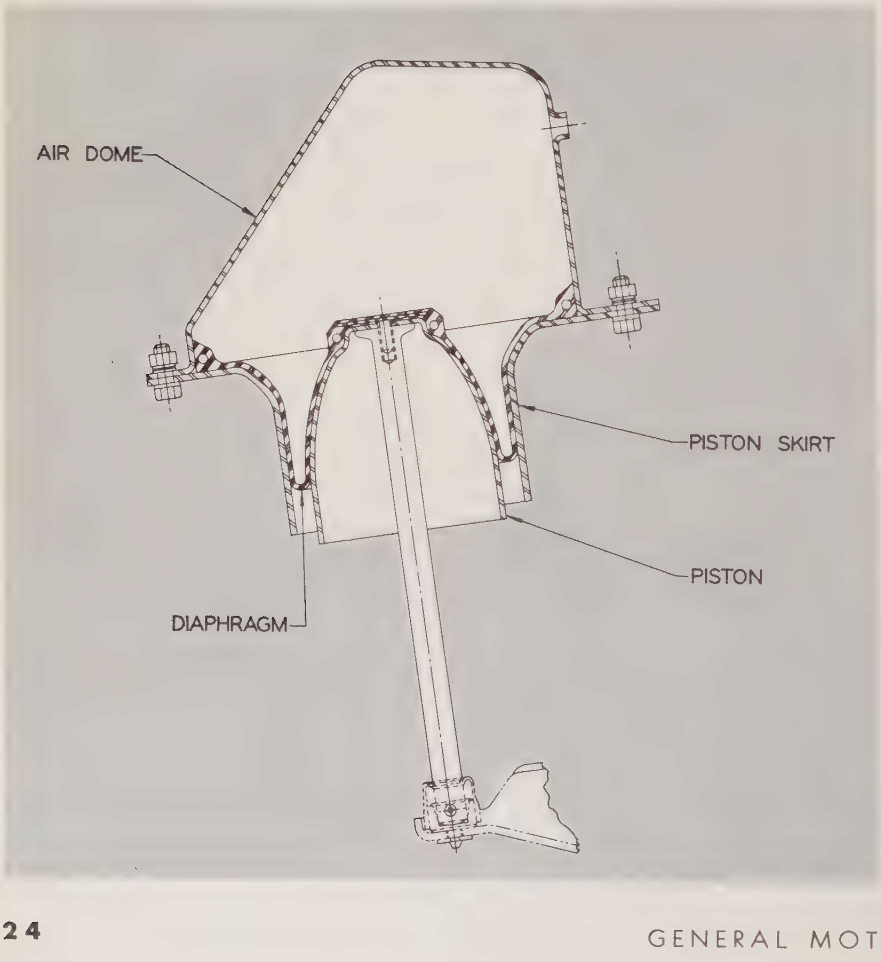
The adaptation of the Engineering Staff's basic design to the Eldorado Brougham naturally brought many changes in detail—for example, the mounting of the air spring in the frame front cross member. One of the design parameters was to retain the basic front suspension arrangement to allow some interchangeability of parts when using either air springs or coil springs. It was, therefore, desirable to use the same front suspension cross member.

Other design parameters were the 75 psi pressure and the spring rate of 55 cycles per minute which would require 450 cu. in. air volume in the air dome of the spring. The first design of piston, diaphragm, and air dome resulted in an air dome too large for the space available in the frame front cross member. This led to a basic redesign of the piston (Fig. 2). It was made with a 3-in. diameter hole at the top. The 126-cu in. volume thus available in the piston reduced the volume required in the dome by 28 per cent. This, in turn, gave a smaller-size air dome which fitted into the space available in the front cross member.

Additional design changes affected (a) the method of attaching the diaphragm to the piston and to the air chamber and (b) the method of attaching the piston rod to the lower control arm. The first design of the piston relied upon air pressure in the dome to hold the inner bead of the diaphragm against the piston. The bottom of the piston was held into a spherical despression in the lower control arm by the same pressure. But experience showed the desirability of having the piston mechanically retained so it could not jump out of position if pressure was lost during car repairs or improper jacking.

In the first designs of the air dome, the dome was welded to the piston skirt. This made it necessary to install the diaphragm through the smaller opening in the skirt and then spring the outer diameter to its proper seat in the air dome. Experience showed that a separate skirt, bolted to the dome after the diaphragm was in position, would be better. This permitted a stiffer wire in the outer bead of the diaphragm and a slight interference between the bead and the dome for better sealing.

Fig. 6 (Left)—The exploded view shows the rear air spring and suspension components which include the spring parts, the leveling valve and actuating link, the lower control arm of the suspension, the upper control arm in the form of a solid yoke, and the shock absorber. The assembled spring is illustrated in the lower diagram. The lower end of the piston rod is rounded and rests in a steel cup attached to an axle bracket. The rod is loosely pinned to the cup to prevent its jumping out of position. A rubber seal around the cup and rod retains a supply of grease to give lifelong lubrication to this connection.



The first skirts were made of fairly light steel. However, laboratory tests revealed deflections in the skirt and the two-piece design made it possible to increase progressively the gage of the steel used and also add reinforcing sections and additional flanges. This change to a separate dome and skirt also was an aid to the Production Department. The assembly of the dome, piston, and diaphragm was simplified. In addition, the bead seat in the dome was exposed and could be better controlled for size and finish.

Durability, Good Sealing Required of Rubber Diaphragm

Selection of the type of rubber diaphragm and the best arrangement for proper sealing against the piston and air chamber were other problems in the design of the air spring components. The quality of the rubber diaphragm from the standpoints of durability and sealing were obviously important in the air spring. Fortunately, the long experience in tire building by the diaphragm supplier aided in its development (Fig. 3). Careful laboratory work, followed by extensive testing under extreme conditions, resulted in diaphragms with a life of over one million cycles on the test apparatus.

A good air seal between the diaphragm bead and the dome and piston is achieved by using a vertical bead seat of a slightly over-size diameter with a stiff bead wire. The bead seats rest against specially-prepared surfaces. During the early experimental work, it was thought that the sealing would be comparable to the sealing of a tire bead to an automotive rim. Tests showed, however, that sealing was not as good because of the different metal forming process used in fabricating the air-spring dome and piston. Tire rims are formed by rolling. The dome and piston are formed by drawing in a die, which leaves a slightly scored metal surface for sealing.

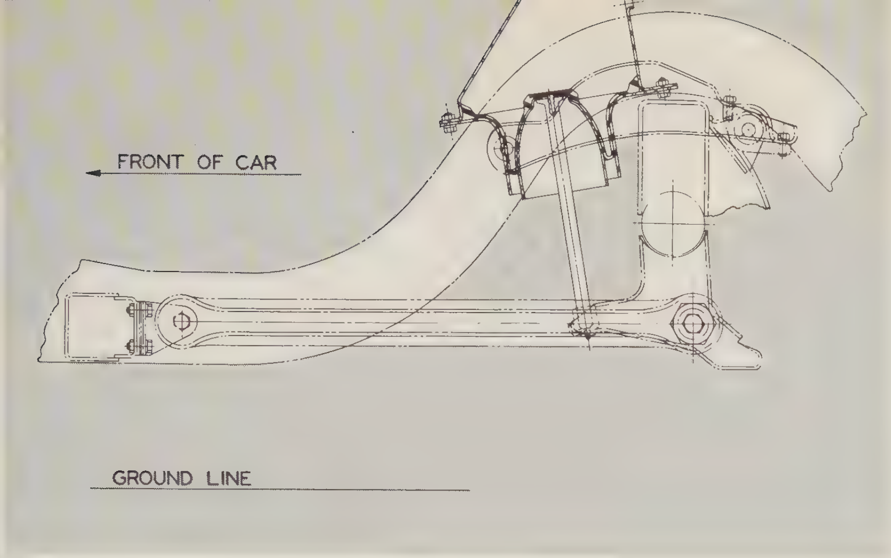


Fig. 7—The rear suspension design is a four-link system. The lower control arms are parallel to the center line of the car and are attached to a frame outrigger and a bracket on the rear axle housing. The upper control arm is a solid yoke which provides a better design for connecting to the new tubular center X-frame. Rubber bushings are used in the connections at the front ends of both the lower and upper control arms. Ball joints are used to attach the rear ends of the control arms to axle housing brackets. This combination of bushings and ball joints on the arms provides durability, freedom in roll, and assures noise isolation.

A test apparatus was devised to evaluate sealing characteristics of different types of metal surfaces (Fig. 4). Many methods of finishing, plating, and sealing were investigated. Results of this testing gave definite and comparable values for the various surfaces and sealants. The following specifications were then adopted:

(a) Use mill-run 1008 cold-rolled, satin-finish steel with surface of 45 micro in. maximum

(b) After forming, buff the sealing areas to remove die marks

(c) Apply a rust-preventative phosphate coat

(d) Apply black primer paint to the same area

(e) Cover sealing areas with silicone grease No. 4.

Continued developmental work resulted in the design of the air spring which was adaptable to the front suspension arrangment used on the standard line of Cadillac cars (Fig. 5).

Rear Air Spring Used with Four-Link Suspension

The rear suspension air springs are similar to those at the front, but the spring parts are not interchangeable (Fig. 6). The piston of the rear air spring is of smaller diameter, as dictated by the air pressure, spring position, and geometry for the four-link rear suspension. The top is closed and has a bullet-nosed shape. A rod is anchored in the piston to connect it to a bracket on the rear axle housing. The shape of the air dome was determined by the volume required (300 cu. in.) and the space available. The method of attachment to the piston skirt is the same as in the front spring. The diaphragm is of the same type as the one used in the front spring, but is not interchangeable (Fig. 3).

As established in one of the initial design parameters, the rear suspension of the Eldorado Brougham is of a four-link type to achieve advantages in lateral stiffness and rear end control. This suspension was designed with lower control arms bolted to the frame and axle housing and with the upper control arm, in the form of a yoke, bolted to the frame and axle housing (Fig. 7). The length of the upper yoke and the lower links, along with the height of their attachments to the frame and to the axle, gives the desired geometry for the proper swing-arm length, rear-end steering, roll center, and rear-end lowering when accelerating. In the actual installation on the chassis, the new suspension system and air springs add only 47 lb to the weight of the car (Fig. 8).

Open-Type Air System Is Solenoid Controlled

The final design problem of the air suspension system for the Eldorado Brougham was to select a method of supplying and controlling the air to produce the expected performance under all operating conditions. The initial design parameters established that the air system would be the *open* type. In this system, the air is compressed, delivered to the springs

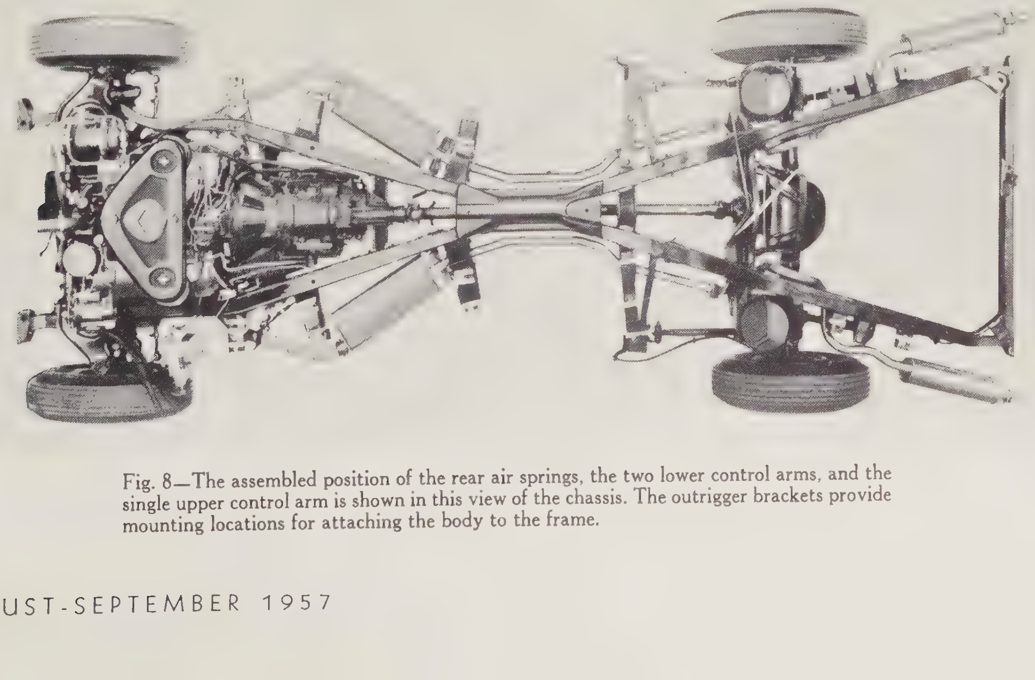


Fig. 8—The assembled position of the rear air springs, the two lower control arms, and the single upper control arm is shown in this view of the chassis. The outrigger brackets provide mounting locations for attaching the body to the frame.

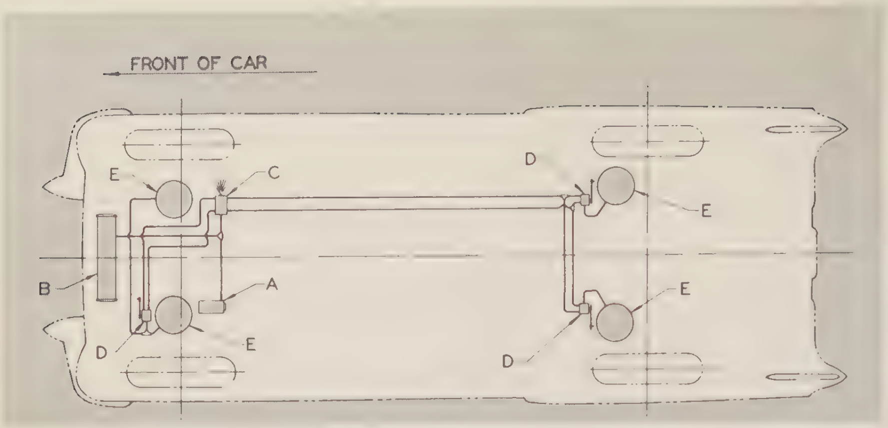


Fig. 9—The basic components of the air supply and leveling system are as follows: compressor *A*; air accumulator *B*; the four-solenoid control unit *C*; wheel leveling valves *D*; air springs *E*. The piping is 3/16-in. diameter copper tubing, which was found to be the minimum size to assure a quick response for leveling.

and then released to the atmosphere after use, eliminating a return system of piping and a low-pressure tank.

One of the problems was to determine the best method of controlling the leveling characteristics under the various conditions of passenger load, road bumps, jacking or shipping, and low temperature. Numerous methods were considered, such as speed-sensitive controls, voltage-sensitive controls, inertia-sensitive controls, pressure-sensitive controls, clock mechanisms or odometer-actuated controls, and delayed-action controls.

The leveling system finally selected combines simplicity and dependability. It consists of a small, electrically-driven air compressor, an air accumulator, leveling valves and arms at the wheels, a set of solenoid controls, and the necessary air piping (Fig. 9). The four solenoid controls, arranged in pairs in a single assembly, meter the flow of high-pressure air to the leveling valves at the car wheels. The first pair of valves allows passage of air only when a car door is open or the ignition is on. Of this pair one valve controls the inlet air to the leveling valves, with the other valve controlling the outlet air. When the car doors are closed and the ignition is off, there is no passage of air and the leveling system is locked out. This provides for all conditions of jacking, shipping, and parking.

The second pair of solenoids control the flow of air under operating conditions. When any door is opened, there is an unrestricted flow of air to the leveling valves. Any change in car height, due to entrance or egress of passengers, is corrected quickly. This occurs with the ignition on or off.

During all driving, there is a restricted flow of air through a coined orifice in the solenoid valve. This permits constant adjustment of car height during running to compensate for loss of weight (as with the consumption of fuel), or for added weight (with an accumulation of mud or ice.) Also, it allows a correction in height when the car is driven away after having been parked on a side hill, or with one wheel on a curb, or in a deep hole. This leveling can be done without excessive use of air.

Severe drops in temperature during over-night parking will, of course, reduce the pressure in the air dome and, consequently, lower the car height. This is corrected as soon as a car door is opened.

Leveling Valves

The leveling valves control the pressure in the air spring at each wheel to keep the horizontal plane of the car parallel to, and at a fixed height above, the ground. The valve is mounted on the frame and connected to the wheel control arm with a mechanical linkage (Fig. 10). Any change in height of the frame actuates the valve, and high-pressure air is admitted to or exhausted from the air spring to restore the normal height of the car.

There are two leveling valves at the rear suspension, one at each air spring. At the front, one leveling valve serves both of the front springs. This arrangement of the three valves establishes the plane for the car and avoids the "fight-effect" that would occur with a leveling valve at each of the four wheels.

The leveling valves are the result of much experimental testing and developmental work by Delco Products Division. They are smaller, lighter, and less complicated than the types that existed at the beginning of the design program.

Each leveling valve has a delay mechanism to prevent leveling action during wheel-hop frequency. This prevents a pumping-up or a bleeding-down of the air spring due to small differences in the inlet and outlet restrictions. Leveling action takes place at all frequencies below approximately 10 cycles per second.

Air Compressor

The air compressor is of the piston-type, having a 0.276-cu. in. displacement and a capacity of 600 cu. in. per minute against a 100 psi head. It is electrically driven and is mounted atop the generator, in the engine compartment. This mounting and location was found best for noise isolation, oil supply, and plumbing. An integral thermal switch protects the motor from overheating. The inlet side of the compressor is piped to the engine air cleaner to insure a clean supply of air.

The compressor operates intermittently and normally runs only about 30 per cent of the car operating time. It is controlled by a pressure switch to maintain a pressure of 120 psi to 130 psi in the air accumulator.

Air Accumulator

A 500-cu in. capacity cylindrical tank, mounted just forward of the radiator top tank, provides a reservoir of high-pressure air which is more than sufficient to level the car from curb to five-passenger load without additional air from the compressor. The tank also serves as a trap for any oil or water and has a manual valve to allow periodic draining of any fluids.

A warning light located on the car instrument panel notifies the driver in case air pressure in the accumulator drops below the required 75 psi during periods of non-operation. This condition is corrected when the engine starts and the compressor operates.

Leak Testing a Vital Step in Assembly

The fabrication and ultimate assembly of the air suspension components were constant considerations throughout the design. At the same time, new methods and procedures were contributed by the Manufacturing and Assembly Departments. As a result of this joint effort, the air suspension system has caused no unusual problems on the production line.

An important manufacturing step was the establishment of a procedure for test-

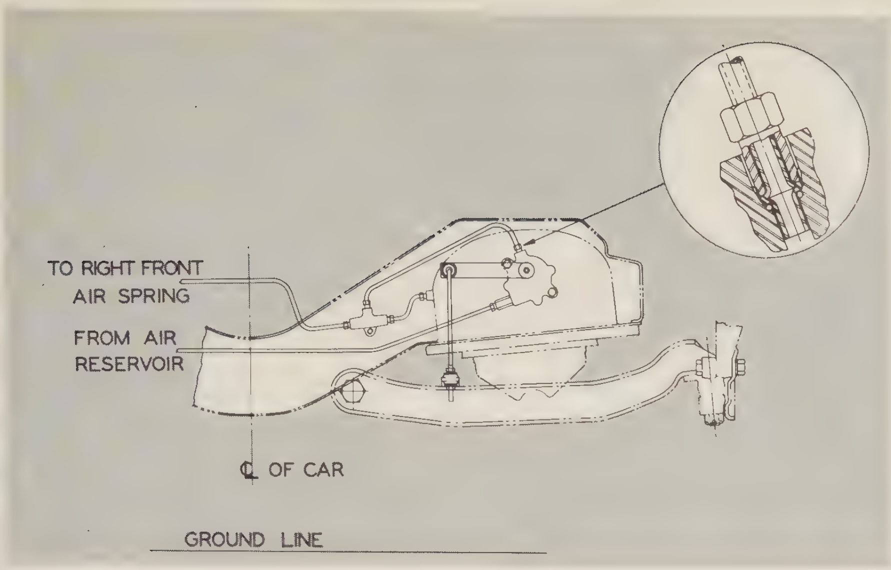


Fig. 10—The arrangement of the leveling valve, control linkage, and air piping is shown in this head-on sectional view of a front air spring. The valve is mounted on the car frame with the actuating linkage connected to the lower control arm of the suspension. Changes in car height cause the valve to either admit or exhaust air, returning the car to normal height. The leveling valve has two tire-type air valves, one for intake and one for exhaust. A single leveling valve controls both front wheel air springs. A tee fitting in the air line connection between the two front springs incorporates a special check valve which permits a fast flow of air during leveling but restrains the flow (through a 0.012 in.-diameter orifice) during roll. Without this restrained flow, the car would have undesirable handling during roll conditions. A typical air line fitting is shown in the enlarged view. To make this connection, a shoulder is upset near the end of the air piping. At assembly, the threaded nut forces the shoulder down against the bottom of the tapped recess in the body of the fitting. This action compresses a rubber ring and assures a leak-free joint.

ing the suspensions to assure a leak-free air system—an obvious requirement in the design. In this procedure, the air dome and front piston are checked for air leaks at 150 psi, under water. The sub-assembly of air dome, piston, piston skirt, and diaphragm is tested at 90 psi with a mixture of four per cent Freon in clean air. A leak detector checks for leaks in the same manner as for air-conditioned car installations. The air dome is first exhausted and then recharged to four psi with pure Freon gas. Repeat checks are then made with the leak detector.

During chassis assembly, an air line is connected to a tire-type valve which holds the air pressure in the dome. When the air-line tube depresses the tire-type valve in the dome, a quick, leak-free connection is made. After the complete assembly of springs, piping, and controls into the chassis, another leak detector test is made, followed by a final check with soapy water on the tubing connections.

Summary

The suspension system was subjected to an extensive testing program after the final design was determined. Tests at Pikes Peak and other high-altitude areas showed that the compressor had adequate capacity for the suspension at any operating altitude. Many thousands of miles on the Belgian block roads at the General Motors Proving Grounds and other road tests have shown satisfactory life for all of the components.

The tests also confirmed that the original goals in the design of the Cadillac air suspension system have been achieved to give a new and better ride with the following qualities:

- The ride is of slower frequency, with slower body motions that are pleasing to the passengers
- The ride characteristics are constant. With the driver alone in the car, or with a full passenger load, plus baggage, there is no change from the designed ride
- The air suspension gives a damping quality that cannot be obtained with conventional springs or with shock absorbers
- The combination of a low center of gravity and the high roll center that is obtained with the four-link rear suspension results in exceptionally good handling of the car
- The constant-height feature improves the appearance of the car, as it is always at show-room height, regardless of the load.

General Motors to Sponsor "Wide Wide World"

General Motors will sponsor NBC-TV's 90 minute Sunday afternoon "Wide Wide World" series during the 1957-58 season. The series will consist of 20 programs which will be telecast every other week beginning September 15. During the previous two years the program was sponsored by a group of General Motors Divisions.

Now in the planning stage are programs which will deal with such areas as the engineering profession, nuclear energy, and America's great inventors. Other areas of science and engineering are tentatively scheduled for future programming.

Among the awards presented to the show for its educational value have been those given by the Thomas Alva Edison Foundation and the Ohio Educational Association.

Each of the "Wide Wide World" shows requires from six to eight weeks of preparation. An average of 60 cameras and 40,000 miles of telephone cable are used to transmit pictures from remote locations back to the master-control switchboard at Radio City in New York. Because the show is telecast live, many complex technical problems regarding transmission of TV waves are introduced. To increase the limited range of TV waves, use is made of the forward-scatter propagation method of transmission.

A Summary of GENERAL MOTORS ENGINEERING JOURNAL Papers Dealing With Various Phases of Automotive Design, Development, and Manufacture

This chart summarizes all papers published to date in the GENERAL MOTORS ENGINEERING JOURNAL dealing with various engineering phases of automotive design, development, and manufacture. Engineering educators whose files may be incomplete may request additional copies of a particular Issue from the Educational Relations Section, Public Relations Staff, GM Technical Center, P.O. Box 177, North End Station, Detroit 2, Michigan. Additional copies of Issues marked with an asterisk (*) are no longer available for distribution.

Solving the Design and Development Problems of Automobile Door Locks
Vol. 1 No. 6

A Typical Problem in Automotive Design: Determine the Shape of a Cantilever Spring of Minimum Stress for Door-Check and Hold-Open Application
Vol. 2 No. 1

Applying the Principle of the Unit-Load to the Packaging of Automotive Hardware
Vol. 2 No. 6

A Step in Body Manufacturing: Processing of Automotive Trim and Hardware for Production
Vol. 4 No. 2

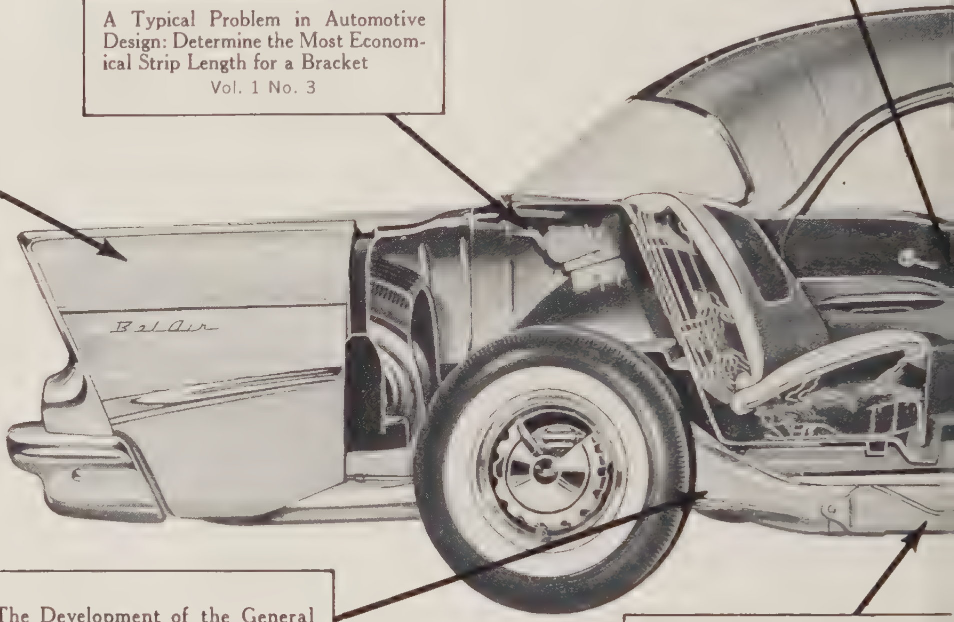
How Stress Problems Are Anticipated and Solved in Automotive Bodies
Vol. 1 No. 7

The Body Draftsman's Function—to Design for Mass Production
Vol. 1 No. 8*

Testing the Structure of An Automobile Body
Vol. 2 No. 3

Weight Control: a Vital Factor in Body Engineering
Vol. 3 No. 5

A Typical Problem in Automotive Design: Determine the Most Economical Strip Length for a Bracket
Vol. 1 No. 3



An Analysis of the Suspension Problems of the Automotive Vehicle
Vol. 1 No. 5

The Interrelationship of Styling and Basic Chassis Design
Vol. 1 No. 6

The Development of the General Motors Air Spring
Vol. 4 No. 3

Applying the Air Spring Design to the Suspension for the Cadillac Eldorado Brougham
Vol. 4 No. 3

The Development of New and Unique Manufacturing Techniques for the Production of Passenger-Car Frames
Vol. 3 No. 2

The Cadillac Tubular Center X-Frame: A New Concept in Automotive Design
Vol. 4 No. 2

Additional Papers of General Interest in the Automotive Design, Development, and Manufacturing Area

Putting a New Car into Production
Vol. 1 No. 2

A Method for Determining the Center of Gravity and Moment of Inertia of an Automobile
Vol. 1 No. 2

The Evolution of a Sports Car: The Chevrolet Corvette
Vol. 1 No. 3

Motorama "Dream Cars" Offer Engineers and Stylists Maximum Design Freedom
Vol. 2 No. 2

How Industrial Suppliers and Design Engineers Work Together
Vol. 2 No. 3

Firebird II: New Gas Turbine Powered "Dream Car" Designed for Highway Use
Vol. 3 No. 2

Engineers Set Serviceability Standards for Reinforced Plastic Automobile Bodies
Vol. 3 No. 5

Applying Physics in Industry: How Studies of Vibration and Noise Aid Developmental Work
Vol. 4 No. 1

A Typical General Motors Institute Laboratory Problem: Find the Horsepower to Give 4 Wheels of an Automobile a Certain Acceleration at a Specified Velocity
Vol. 4 No. 2

Industrial Engineers at Work: Some Typical Processing Problems Resulting from an Annual Model Change
Vol. 4 No. 2

cal Problem in Automotive Design: De-
the Length of a Shift-Rod
Vol. 1 No. 1

Problems of Combination Signal Seeking
sh Button Radio Receivers
Vol. 1 No. 1

ysics of Automotive Radio-Interference
tion
Vol. 1 No. 5

cal Problem in Automotive Design: Plot
und Plans of Visibility from the Driver's
in an Automobile
Vol. 1 No. 5

al Engineering Moves Alongside Mechan-
gineering in New Speedometer Design
Vol. 1 No. 5

nary of Major Developments in the Steer-
chanisms of American Automobiles
Vol. 2 No. 2

cal Problem in Engineering: Determine
gular Position of Two Punched Holes to
ly Balance a Speedometer Speed Cup
Vol. 2 No. 6

Engine-Transmission Relationship for Higher Efficiency
Vol. 1 No. 1

A Discussion of 12-Volt Automotive Electrical Systems
Vol. 1 No. 1

A Study in Applied Physics: Locating the Piston Pin to Minimize Piston Slap
Vol. 1 No. 3

The Buick Air Power Carburetor
Vol. 1 No. 4

The Effect of Combustion-Chamber Shape on Smoothness of Power
Vol. 1 No. 4

General Motors Automotive Engine Test Code
Vol. 1 No. 5

Designing Simplicity, Ease of Manufacture, and Safety into an Automotive Electrical System
Vol. 1 No. 5

The Design, Development, and Manufacture of Hydraulic Valve Lifters
Vol. 1 No. 6

Applying Radiochemistry to the Manufacture of Oil-Filter Elements
Vol. 1 No. 8*

General Motors Test 20—a New Standard Means for Determining Gross Horsepower
Vol. 1 No. 9*

A Discussion of Economic Factors Affecting the Steel Selection and Heat Treatment of Automotive Gears
Vol. 2 No. 5

Progressive Mechanization as Applied to Core Making in a Production Foundry
Vol. 3 No. 1

A Study in the Design of Sand Molded Engine Castings
Vol. 3 No. 2

The Development of a High-Output Carbon Pile-Type Generator-Regulator
Vol. 3 No. 2

The Application of Radiation Techniques to Engine Combustion Studies
Vol. 3 No. 3

Research Studies on Automotive Engine Fuel Economy
Vol. 3 No. 3

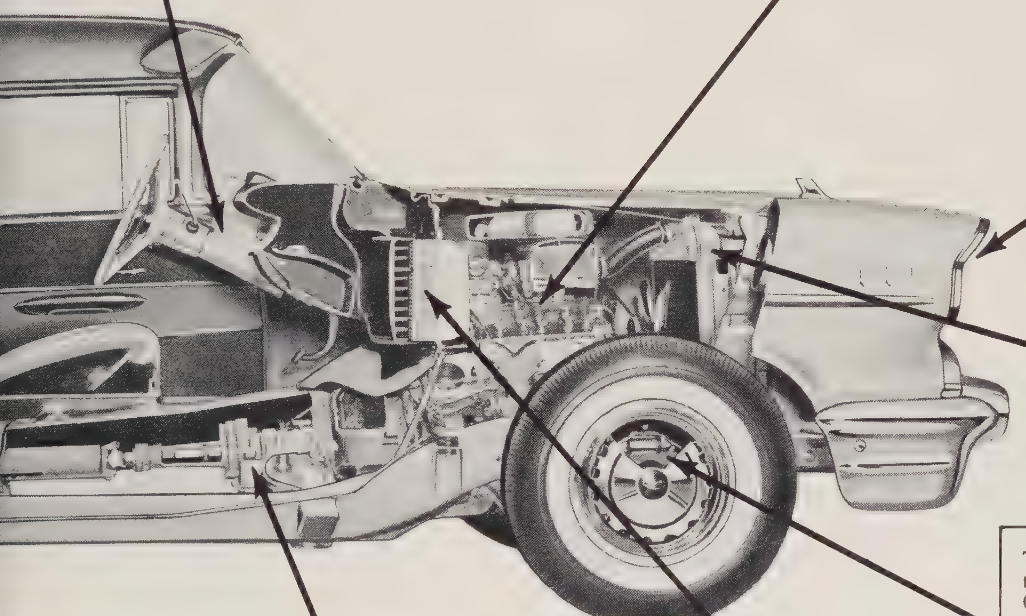
Bench Test Simplifies Search for Better Cam and Tappet Materials
Vol. 3 No. 4*

New Test Facility Provides High-Capacity Air Flow for Automotive Air Cleaner Development
Vol. 3 No. 4*

Matching Compression Ratio and Spark Advance to Engine Octane Requirements
Vol. 3 No. 5

A Discussion of Design Factors for a 12-Volt, 4-Pole, Wave-Wound Automotive Engine Cranking Motor
Vol. 4 No. 2

A Discussion of the Basic Design and Operation of the General Motors Fuel Injection System
Vol. 4 No. 3



How the Autronic-Eye Automatic Headlamp Control was Designed to Meet Driver Requirements
Vol. 1 No. 2

Major Cooling Problems Encountered in Today's Automobiles
Vol. 1 No. 4

The Engineer and the Cooling-System Thermostat: A Story of Evolution
Vol. 1 No. 9*

Solving the Thermal and Structural Problems in Radiator Design for Automotive Cooling Systems
Vol. 2 No. 4

The Chemical and Physical Properties of Brake Fluids for Safe Vehicle Operation
Vol. 1 No. 6

The Theoretical and Practical Aspects of Automotive Brake Design and Testing
Vol. 2 No. 5

A Typical Problem in Automotive Design: Determining Exhaust Valve Cone Angle for a Ball in the Dynaflow Transmission
Vol. 1 No. 4

A Typical Problem in Automotive Design: Determine the Tensile Stress in Pump-Cover Bolts on a Torque-Converter Transmission
Vol. 1 No. 6

An Application of Hydraulic Fundamentals: Development of the Hydra-Matic Automatic Transmission
Vol. 1 No. 7

Operating Principles of Buick's Twin-Turbine Dynaflow Torque Converter
Vol. 1 No. 8*

A Typical Problem in Automotive Design: Determine the Length of the Governor Valve in the Hydra-Matic Automatic Transmission
Vol. 1 No. 8*

A Study of Applied Gear Design: Speedometer Gears of Different Ratios in Pontiac Transmissions
Vol. 2 No. 1

The Manufacture of Planet Pinions
Vol. 2 No. 6

Operating Principles and Applications of the Fluid Coupling and Torque Converter to Automatic Transmissions
Vol. 3 No. 3

A Typical Problem in Engineering: Determine the Design Specifications for a Pressure Regulator Coil Spring
Vol. 4 No. 2

Development of the Cadillac Air Conditioner
Vol. 1 No. 1

Development of an Automobile Air Conditioning System for Underhood Installation
Vol. 2 No. 3

The Frigidaire Package-Type Air Conditioning Unit
Vol. 2 No. 4

A Product Engineering Study: Design of an Axial-Type Refrigeration Compressor
Vol. 3 No. 3

Preparation and Evaluation of an Industrial Report

By CHARLES L. TUTT, JR.
General Motors Institute

Industrial reports assist management in making policy and operating decisions by presenting factual information about current or future operating and product design problems. Because of their importance it is essential that such reports be prepared in a uniform manner and be complete, concise, and accurate. Since 1945 General Motors Institute has supervised the preparation of over 2,000 industrial reports required as part of the Fifth-Year program leading to a Baccalaureate degree in either engineering or business administration. Experience gained with the wide range of material covered in these reports has led to the development of specifications which maintain organizational uniformity in their preparation and final make-up. A set of guiding questions also has been developed which serves to evaluate the report as to its technical content and logical presentation of this content.

Report value: a matter of presentation as well as material content

INDUSTRIAL reports are a tool of management. They act as a guide to policy or operating decisions by presenting factual information about current or future operating and product problems. Subjects of industrial reports vary from non-engineering areas, such as accounting, traffic, and personnel management, to technical areas of manufacturing operations and product engineering.

Industrial reports supply management with information such as whether it is feasible to apply automatic data processing methods in an accounting department, whether a proposed change in a manufacturing process will prove economical, or whether a proposed change in the design of a product is feasible.

Because of the broad nature of information supplied management by the industrial report any engineer or non-engineer member of an industrial organization may be required at one time or another to prepare and write a report. The reports may vary in length from a single typewritten page to hundreds of pages bound in book form. Each report, no matter the length, serves an important purpose.

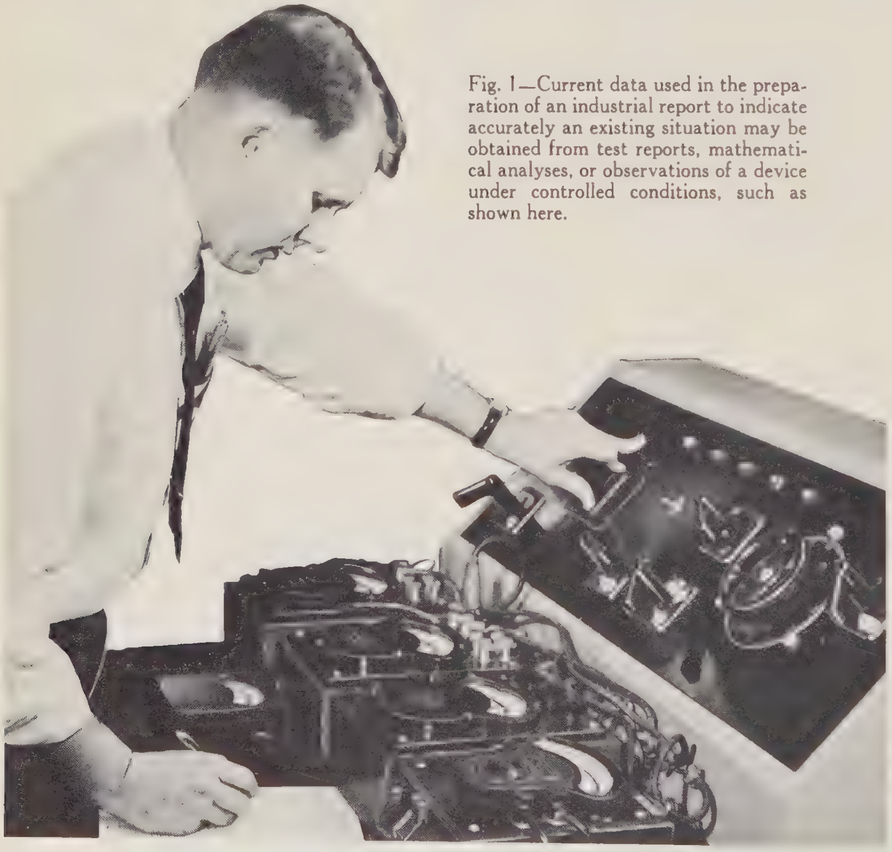


Fig. 1—Current data used in the preparation of an industrial report to indicate accurately an existing situation may be obtained from test reports, mathematical analyses, or observations of a device under controlled conditions, such as shown here.

Value of Report Depends on a Clear Objective and Writer's Ability to Find, Analyze, and Appraise Data

Industrial reports are prepared with a specific objective in mind. The objective should:

- Define clearly the problem and indicate its scope
- Indicate the purpose and/or need of the report.

Once the objective of a report is set it then becomes the responsibility of the writer to determine how and where to obtain data necessary to prepare the report and to substantiate findings, recommendations, or problem solution. Experience has shown that there are two types of data:

(a) Data resulting from studies, analyses, and reports which have been completed prior to the report assignment

(b) Data which must be obtained to indicate the existing situation as accurately as possible.

Few industrial reports are based on data obtained from published material found in periodicals, books, or manufacturer's catalogs. The writer must depend, therefore, on locating information pertinent to the assigned report in the industrial organization of which he is a part.

Members of an industrial organization

constantly make studies and collect data to assist in meeting daily responsibilities and in making decisions. Thus, a great amount of data in the form of memos; research, test, and departmental reports; and an individual's memory of how similar problems have been handled in the past provide a good background as to where to find and how to collect the necessary data.

Current data can be obtained in many ways. In a technical report, data can come from such sources as test reports, observations of a device under controlled conditions (Fig. 1), or a mathematical analysis. Non-technical reports must depend on such devices as surveys, questionnaires, or statistical analyses.

The value of an industrial report depends upon the validity, accuracy, and soundness of the data which supports any conclusions, findings, or problem solutions. The value of the data, in turn, is dependent upon the time spent in preparing and setting specifications as to how it must be obtained. It is essential, therefore, that the writer of an industrial report have:

- Knowledge of how similar problems have been handled in the past
- Ability to acquire and appraise data
- Knowledge of informational sources both within and outside the organization
- Ability to subject the problem to critical and analytical thinking
- Ability to put together fact and reasoning into readable and understandable English.

Over 2,000 Industrial Reports Prepared by G.M.I. Graduates

During the past 12 years, General Motors Institute has been working with GM Divisions and Staffs in the preparation of a wide range of industrial reports written by graduates of the 4-year co-operative programs. Before being awarded an appropriate Baccalaureate degree in either engineering or business administration a graduate of a 4-year co-operative program must submit evidence to G.M.I. that he can apply the knowledge and experience gained during his program studies and work experience. This is achieved through a Fifth-Year Project Study Report program in which a student is assigned to solve a problem which is of current or future interest to the sponsoring Division or Staff and acceptable to G.M.I. The graduate is guided while solving the assigned problem by both an Institute faculty member and a technical advisor appointed by the sponsoring Division or Staff. A Baccalaureate degree is awarded when the graduate meets the objectives of the assigned problem and the written Project Study Report is accepted by both the sponsoring Division or Staff and G.M.I.

During the period that the Project Study Report program has been in operation over 2,000 industrial reports have been written. Of these, over 60 per cent have concerned manufacturing line or staff operations and approximately 20 per cent have been in the product engineering areas of design, development, testing, and research. The remaining reports have dealt with areas of sales and service, personnel management, accounting, traffic, purchasing, and material control.

Specifications Developed for Each Part of Report Assure Uniform Presentation

Over the 12-year period that the Fifth-Year Project Study Report program has been in operation, G.M.I. has found it necessary to develop specifications which not only cover the wide range of subject matter but also maintain a uniformity in the preparation of these reports. These specifications, based on experience, recognize that the industrial report should have four basic parts presented in the following order: (a) introduction, (b) conclusions (recommendations or findings), (c) main discussion, and (d) appendix.

A preface should be included if required. Also, the use of a table of contents and list of illustrations aids in quick and efficient use of the report.

The following is a discussion of G.M.I. specifications as applied to each part of an industrial report.

The Preface

A preface should include pertinent information concerning the situation that led to the report assignment and necessary acknowledgements. The industrial plant situation is explained first so that the significance of the report will be more apparent to the readers. Acknowledgements should be made to those who contributed substantially to the report or success of the problem solution. The nature of each contribution should be explained.

Fig. 2—The author of an industrial report should use a style of writing which is objective, impersonal, unemotional, and unbiased and should present information in an orderly and accurate manner.

The Introduction

The introduction of an industrial report should clearly and immediately state the problem which existed, as well as the exact purpose or objective of the report in relation to the problem. Background material necessary to understand the report and investigation may be included, but should be held to a minimum.

The procedure of investigation is part of the introduction and should give a clear and concise picture of the method used. This enables the reader to judge the validity of the report. In reports where there is no step-by-step procedure of conducting the investigation, the introduction should end with a statement indicating the order of discussion in the remainder of the report.

Background information included in the introduction should be limited to that needed to orientate the reader. This includes necessary definitions, history, or other such pertinent information.

The Conclusions (Recommendations or Findings)

The introduction should be followed by the conclusions and/or recommendations, if any are to be made. Where no conclusions or recommendations are made, a summary of findings is desirable.

General conclusions or recommendations should precede those which are specific. Each specific conclusion, recommendation, or finding should be keyed to the substantiating data in the main discussion of the report. Page references may be made or the conclusions, recommendations, or findings may be grouped under headings corresponding to the sections making up the main discussion of the report. In any case, a listing of the conclusions, recommendations, or findings is recommended.

The Main Discussion

The sections of the report which make up the main discussion should be organized to read independently. The opening sentence of each section should make clear that part of the report to which the section is related. If a section deals with only one phase of the investigation, a detailed procedure can be described either immediately following the opening sentence or at the most appropriate point in the discussion. Also, the opening of each section should explain the organization of data which are to be included.

Data must be complete, accurate, specific, and conclusive. The conclusions, recommendations, or findings, are valid only to the extent that they are supported by data obtained through a logical procedure of investigation. The data and procedure used should be presented so that the reader arrives at the same conclusions as the writer. The discussion of the data should be concluded with a summary of the conclusions, recommendations, or findings of the section.

In cases where a section is primarily informational, its organization necessarily will vary from one which is investigational. For example, if a section reports an analysis of a method used in a manufacturing plant, the opening should indicate the purpose of the section, present briefly the necessary background material, and give the order of discussion of data. At the end of the section, the data should be summarized.

The Appendix

An appendix is often needed to present detailed data, background, or other information not required for the development of the main discussion of the report. Material included in the appendix is required to give added support to the conclusions, recommendations, or findings through such means as photographs, drawings, graphical presentations, calculations, or tabulations.

The above specifications allow the writer considerable freedom as to style of writing and use of illustrative material (Fig. 2). The style of an industrial report, however, should be objective, impersonal, unemotional, and unbiased. Report content should be written so that it is direct, sound, accurate, orderly, and precise. The language should be that of the writer, as he is personally responsible for the report.

Evaluation of the Report

Technical Evaluation	Report Evaluation
Total Report	
Does total report: • Fulfill original objectives? • Cover all points essential to the main purpose of the assigned report? Is each part of the report in proportion to the complexity or significance of the corresponding parts of the assigned report?	Does total report have: • Information arranged and presented in a sound and effective manner? • A suitable title page giving essential information? • A table of contents and list of illustrations which are complete, accurate, and easy to understand?
Preface	
Does preface: • Explain the assigned report accurately? • Explain any change from the original objective during the development of the report? • Give an accurate account of the work experience related to the assigned report? • Acknowledge the help of those who contributed to the preparation of the report?	Does preface: • Explain the reasons which gave rise to the selection of the report assignment? • Give a clear account of work experience relating to the assigned report? • Make acknowledgements in proper form: name, title and type of aid?
Introduction	
Does the introduction: • State the problem so that it corresponds with the objectives? • Give the main steps of the procedure accurately?	Does the introduction: • Give enough background information so that the nature of the report is easily understood? • State the problem clearly and without ambiguity? • Make clear the main steps in the procedure? • State the specific purpose of the report? • Present the order of discussion for the remainder of the report?
Conclusions	
Are conclusions: • Significant, pertinent, and valid? • Substantiated by the data presented in the report? • Appropriate to the assigned report? • Supplemented by a summary of findings and recommendations when necessary?	Are conclusions: • Identified by an opening statement? • Grouped with general before specific conclusions? • Presented by including page numbers of substantiating text or grouped according to sections of the text so that the reader can locate substantiating data for any statement? • Accurately grouped and labeled as conclusions, recommendations, or findings? • Presented in lists, charts, graphs, or other suitable means of summarizing?

Table I—These guidelines for the evaluation of a report were developed at General Motors Institute. The guidelines are designed to assure uniform evaluation and to maintain a high uniform standard of quality.

Evaluation of the Report (Continued)

Technical Evaluation	Report Evaluation
Main Discussion	
Does main discussion: • Describe any tests, equipment, or procedures used with sufficient completeness and accuracy so that the reader could use the information as a guide to a similar problem? • Give specific sources of all information? • Give complete and exact figures for all results and omit inexact information such as "many," "few," and "more?" • Use technical terminology accurately?	Does main discussion: • Identify the area to be considered? • Indicate the order in which each section is arranged? • Explain clearly how data were gathered, processed, and interpreted? • Relate data clearly to the conclusions reached? • Give exact source of information in proper form, including footnotes and bibliography when needed? • Use precise language? • Develop the discussion in a logical and clear order so that the material is easily usable? • End with a summary or grouping of conclusions developed in the section?
Illustrations	
Are illustrations: • Provided when necessary as determined by the nature of the data or discussion? • Consistent in quality with the text discussion? • Accurate in their interpretation? • Sound in the techniques used?	Are illustrations: • Referred to in the text? • Placed near text reference? • Reproduced properly? • Clearly numbered and labeled? • Used when desirable as a summary device?
Appendix	
Does the appendix: • Contain essential supporting or supplementary material? • Contain all material which belongs there?	Does the appendix: • Contain appropriate material? • Have a usable arrangement?
English	
	Is the English: • Accurate in following the conventions of language? • Clear, precise, and readable?

Guidelines Established to Evaluate the Industrial Report

Once an industrial report is completed, the important job of evaluating it begins. An evaluation should be made to determine first the completeness and accuracy of the content and then to evaluate the report as a clearly and logically organized presentation of the development of the content.

At General Motors Institute, obtaining uniformity in evaluating some 300 reports a year by 90 faculty members became a major problem when attempting to maintain a consistently high level of quality in all reports. It was necessary, therefore, to establish guidelines to assist the readers in obtaining a uniform evaluation of these reports (Table I). These guidelines, in the form of questions, cover each part of the report and are designed to evaluate the report as to content, preparation, and organization.

Nature of Report Determines Reproduction Method to Use

Experience has indicated that the nature of the report determines the number to be reproduced. In turn, the number required generally determines the method used for reproduction.

A typewritten original with carbon copies will meet the need when four or less copies are required. This method, however, does not allow for additional copies if required later. Typed vellum masters with a carbon backing will produce an unlimited number of copies by any one of the ammonia blueprint processes but are not economical for runs of more than eight or ten copies. Additional, individual copies, however, can be reproduced from the vellum masters whenever required. The most favored process for reproducing G.M.I. reports at the present time is to prepare paper plate masters and reproduce them by the multilith process. Runs of from 10 to 100 copies are economical by this method. Additional copies are reproduced easily from the masters.

To increase the use of industrial reports within an organization, a subject listing of report titles with a brief abstract of each report is of value. The abstract should be short (50 to 100 words) and should enable a person to determine whether reading the full report will be of value. The abstract should give a clear indication of the objective and scope of the report, mention the procedure, and indicate the results achieved.

Conclusion

The industrial report is used not only to increase available information on a subject or problem but also to communicate that information to management and personnel involved. Today, increasing emphasis is placed upon the use, recording, and availability of industrial reports. Their value increases still further as their availability increases within an organization. It has been the experience of General Motors Institute that uniformity of organization and quality of preparation gives a systematic approach to industrial report writing and results in their greater use.

Why Can't the English . . .

By THEODORE L. CHISHOLM
Patent Section
Central Office Staff

The patent attorney's English must define the scope of the invention

"WHY can't the English," wails one of the characters in *My Fair Lady*, "learn to speak?"

"Why can't the patent attorneys," complain the engineers, "learn to write plain English? Why do they use weird terms for everyday mechanical things and tangle these terms into a fantastic obscurity of language?"

The engineers have a point. While many patents are clear and easy to read, it is true that many are inexcusably difficult. One of the reasons for this is plain poor writing. There are, however, other patents which are well written but annoy engineers and judges with their unfamiliar and difficult terms. These people want to know, for example, why the term "translating device" is used to refer to a plain electric light bulb, or why a simple clutch is referred to as a means responsive to an increase of fluid pressure adapted to establish drive through a transmission. This requires explanation.

Patent Claims Must Separate the Old from the New

In writing the claims of a patent, the patent attorney is writing for the inventor—the inventor's declaration of what is his under the invention and what is not. This declaration binds the inventor for the next 17 years. Thus, the patent attorney must write a definition of the thing invented which will do all of the following:

(a) it must exclude everything done before—that is, distinguish from the prior art

(b) it must include the essential elements of the invention—that is, enumerate all parts and relationships without which the invention would not work

(c) it must *not* include non-essential parts or relationships, for the wording of the claim should represent the inventor's statement of what his invention is. He stands or falls by that statement. If an unnecessary part has been included in this statement, a competitor can often use the invention, leave out this unnecessary part, and avoid the patent entirely

(d) it should include all conceivable equivalents of each individual element that is included in the invention because, while the law attempts inherently to give each inventor a reasonable range of equivalents, there can be expensive arguments of uncertain outcome as to whether a certain thing is an equivalent.

This last is a large and difficult responsibility. Attempting to carry it out frequently results in language which engineers do not appreciate and judges do not understand, but which is both proper and necessary.

Patent Nomenclature Used for Maximum Benefit and Protection of Inventor

Suppose the patent attorney is writing a patent application on an invention for controlling the rate of flow of oil into a clutch for an automatic transmission. Suppose it is necessary to include the clutch as an element of the claim, but the essence of the invention is in the apparatus for controlling the flow, in combination with the clutch. There are reasons why the patent attorney cannot just say "a clutch."

Basically, what the clutch does is selectively make or break the drive train through the transmission. Many forms of device can do this satisfactorily for this purpose. A simple, traditional disk clutch connecting an input to an output shaft will do. So will a fluid drive—that is, a pump and a turbine which make the drive through the transmission when they are filled with oil and break it when empty. But is this properly termed a clutch? This question has been debated extensively. Likewise, the drive can be made by having one side of the disk clutch grounded and the other side connected to a reaction gear of a planetary gear set to stop or hold the reaction gear. Is this clutch still a clutch? Engineers call it a clutch, patent attorneys think it is a clutch, but the Patent Office says it is not. The Patent Office says it is not to be called a clutch but it can be called a brake.

This results from a Patent Office rule based on its definition of a clutch as invariably joining together two rotating parts and its definition of a brake as invariably having one stationary part. No clutch can ever be a brake and vice versa. So the Patent Office requires attorneys to distinguish and elect between clutch and brake.

In this hypothetical case, it makes no difference which is involved. An obvious suggestion is to say "clutch or brake." But, this runs afoul of a second Patent Office rule against using alternative expressions in claims. So the patent attorney must say something which means either or both and yet complies with Patent Office rules. If there were some recognized word like "brutch" or "clake," it would be simple. Since there is not, the patent attorney must find a term or expression which means any form of clutch, brake, or equivalent device which selectively establishes drive through a transmission and breaks the drive.

The patent attorney is now driven to an approach inherent in basic English and in Pacific pidgin English. The clutch or brake becomes a thing that is used to make or break drive through a transmission when it is pumped full of oil under pressure. This is reminiscent of pidgin English for piano; namely, "big-fella-box—you-fight-him-he-cry." The patent attorney dignifies this into "means responsive to an increase of fluid pressure adapted selectively to establish and interrupt torque in the transmission."

So much for the *claims*. In those parts of the *description*, which are illustrative

Notes About Inventions and Inventors

Contributed by
Patent Section
Central Office Staff

THE following is a general listing of patents granted in the names of General Motors employes during the period September 1, 1956 to December 31, 1956.

AC Spark Plug Division
Flint, Michigan

• **Thomas E. Jacobi,** (*B.E.E., Marquette University, 1942*) experimental engineer, Missile Guidance System Engineering Department, inventor in patent 2,769,125 for a multi-speed synchro data transmission system.

• **Roy L. Bowers,** (*B.S., Michigan State University, 1930*) staff engineer, Engineering Department, inventor in patent 2,772,001 for a filter for fluids, such as gasoline.

• **Thomas E. Weiher,** (*B.S.E.E., Marquette University, 1950*) senior research engineer, Regulus Missile Engineering Department, inventor in patent 2,772,412 for synchro testing.

• **Warner E. Remington,** chief engineer, Manufacturing Development, inventor in patent 2,772,656 for a filter impregnation injection machine.

Allison Division
Indianapolis, Indiana

• **Carson O. Donley,** (*B.S.M.E., University of Cincinnati, 1939*) now chief engineer, Engineering Department, Diesel Equipment Division, inventor in patent 2,765,-520 for a bearing and method of making same.

• **William F. Egbert,** (*B.S.Aero.E., Tri-State College, 1941*) group head, Exhaust and Fabricated Sections, Turbo-Jet Engineering Department, inventor in patent 2,765,620 for a flow deflector for a combustion chamber apparatus.

• **R. Donald Tyler,** (*B.S.E.E., Rose Polytechnic Institute, 1947, and B.S. in business administration, Butler University, 1955*) superintendent of electronics development, Electronics and Parts Test Department, and **John M. Whitmore,** (*B.E.E., The Ohio State University, 1936*) head, Electronics and Parts Test Department, inventors in patent 2,766,617 for a torquemeter.

• **Roy H. Brandes,** project engineer, Engineering Department, Aeroproducts Operations, and **Robert C. Hamilton** and **Richard E. Moore,** no longer with the Division, inventors in patent 2,766,731 for an electrically controlled fluid pressure-operated remote positioner.

• **Ned W. Lowry,** (*B.S.E.E., University of Dayton, 1949*) senior project engineer, Engineering Department, Aeroproducts Operations, and **Robert K. Skinner,** (*B.S. E.E., University of Cincinnati, 1943*) senior project engineer, Design Engineering Department, Aeroproducts Operations, inventors in patent 2,766,833 for a propeller control system.

• **Donald G. Zimmerman,** (*B.S.M.E., Purdue University, 1940*) section chief on design projects, Power Turbine Engineering Department, inventor in patent 2,766,963 for a turbine stator assembly.

• **Arthur W. Gaubatz,** (*B.S., University of Wisconsin, 1920*) senior project engineer, Experimental Engineering Department, inventor in patents 2,767,275 for a differential pressure switch and 2,778,241 for a discriminator device.

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only and do not state the claim nor mark out the breadth of the invention, it is not necessary to the protection sought to use such clumsy terms. But this nomenclature frequently is carried through the entire patent description by momentum or through force of habit in writing. This does make difficult reading. It may even be dangerous—for example, when a judge responsible for deciding an infringement suit does not understand the patent nomenclature or is annoyed with it.

The Goal: to Help a Layman Understand a Precise, Legal Document

If, however, the patent attorney uses plain simple terms, like clutch, in the *description* and then, without transition, jumps to pidgin English in the claims, the engineer and the judge justifiably wonder what the claims mean. Many litigants, having contradictory opinions, have hired expensive patent experts to explain to the court their contentions as to the meaning of a claim, or even a single term.

This gets the careful patent attorney into the desirable habit of having his language in the description bear some recognizable resemblance to the wording of his claims. This can be done by making the description its own translation of the claims so that judge, engineer, and lawyer can understand every term in the patent. Suppose the drawing, which illustrates merely one form of the invention and is part of the patent, shows a clutch identified by a reference character 19. The description might be worded: "The clutch 19 connects shafts 20 and 21 when its actuating chamber 22 is filled. This is illustrative of any suitable device which can selectively establish and interrupt drive through the transmission when supplied with oil under pressure and emptied, respectively."

Thus, the selective use of the English language can provide a bridge from the engineer's clutch to the patent attorney's legal language in that part of the patent which is analogous to a deed.

• **Charles F. Hayes**, (*General Motors Institute, 1943*) head of power turbine combustion, Turbine Fuel Systems and Components Engineering Department, inventor in patent 2,768,497 for a combustion chamber with swirler.

• **Robert J. Wente**, (*B.S.M.E., Purdue University, 1941*) head of advanced design power turbine controls, Aircraft Engine Engineering Department, **Edmund M. Irwin**, (*B.S. in physics, Central Michigan College, 1938*) head of control electronics, Fuel Systems and Components Department, **Floyd J. Boyer**, (*B.S.M.E., Georgia School of Technology, 1940*) group project engineer in turbo-prop engine controls, Engineering Department, and **Thompson Baber**, no longer with the Division, inventors in patent 2,768,504 for a dual engine fuel system.

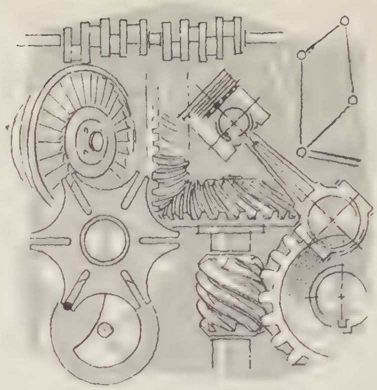
• **Walter D. Eby**, (*B.S.M.E., Purdue University, 1942*) senior project engineer, Production Engineering Department, Aeroproducts Operations, and **Harold H. Detamore**, no longer with the Division, inventors in patent 2,768,612 for a multiple piston servomotor.

• **Howard M. Geyer**, (*B.S.I.E., University of Alabama, 1940*) chief product research engineer, Engineering Department, Aeroproducts Operations, inventor in patents 2,769,430 for an actuator with dual locking means, 2,773,485 for a self-locking actuator, 2,774,336 for a locking means for a fluid motor or the like, and 2,774,337 for an actuator assembly with load-sensing means.

• **Edgar G. Davis**, (*General Motors Institute, 1935*) manager, Bearing Department, inventor in patent 2,770,586 for a grid bearing and method of making same.

• **Chester E. Hockert**, (*B.S.M.E., Armour Institute, 1937, and M.S.M.E., Cornell University, 1941*) chief engineer, Turbo-Jet Engineering Department, and **Leslie R. Smith**, (*International Correspondence School*) chief draftsman, Turbo-Jet Engineering Department, inventors in patent 2,772,069 for a segmented stator ring assembly.

• **Kenneth L. Berninger**, (*Purdue University*) senior project engineer, Engineering Department, Aeroproducts Operations, and **William A. Weis**, (*B.M.E., University of Dayton, 1938*) senior designer, Engineering Department, Aeroproducts



Operations, inventors in patent 2,773,452 for a pump control system.

• **John B. Wheatley**, (*A.B.M.E., 1929, and M.E. in aeronautics, 1930, Stanford University*) assistant chief engineer, Advanced Design Engineering Department, inventor in patents 2,773,667 for a turbine rotor sealing ring and 2,779,531 for a gas turbine engine with hydraulic thrust balancing.

• **William S. Castle**, (*B.S., University of Illinois, 1942*) group leader on turbines and compressors, Jet Engine Design Engineering Department, and **Chester E. Hockert*** inventors in patent 2,773,711 for a coil spring fluid pressure seal.

• **Victor W. Peterson**, (*B.S.M.E., Rose Polytechnic Institute, 1939*) engineer, Advanced Design and Development Department, inventor in patent 2,775,331 for a clutch with coolant metering.

Brown-Lipe-Chapin Division
Syracuse, New York

• **Stanford Landell**, (*University of Pennsylvania*) works manager, inventor in patent 2,772,924 for a wheel cover.

Buick Motor Division
Flint, Michigan

• **Joseph D. Turlay**, (*B.S.M.E., Oregon State College, 1928*) staff engineer, Advanced Engineering, inventor in patent 2,765,781 for a valve actuating mechanism.

• **Harry C. Doane**, now assistant to the vice president in charge of GM Engineering Staff, inventor in patents 2,770,971 for a flywheel, clutch, and starter housing drain and 2,779,164 for a refrigerating apparatus for an automobile.

Cadillac Motor Car Division
Detroit, Michigan

• **Philip W. Maurer**, (*General Motors Institute, 1934*) staff engineer—electrical and accessories, Engineering Department, inventor in patent 2,779,154 for a control means for clocks and the like.

Chevrolet Motor Division
Detroit, Michigan

• **Maurice S. Rosenberger**, (*Nebraska Wesleyan University*) assistant chief engineer in charge of engine and passenger car chassis design, Engineering Department, inventor in patent 2,766,639 for a transmission control system.

• **W. S. Wolfram**, (*B.S. Auto. E., University of Michigan, 1932*) assistant staff engineer, Research and Development Department, inventor in patent 2,768,536 for a transverse engine and transmission.

• **Marvin T. Wobrock**, (*Wayne State University and University of Michigan*) senior project engineer, Truck Chassis Group, Engineering Department, inventor in patent 2,770,341 for a clutch assembly.

• **A. E. Kolbe**, future engine design engineer, Engine Design Department, inventor in patent 2,771,865 for a manifold support structure.

Cleveland Diesel Engine Division
Cleveland, Ohio

• **Carl A. Bierlein**, (*General Motors Institute, 1933, B.M.E., 1948*) director, Inspection and Test Department, and **George W. Bixler**, (*B.S.M.E., 1933, and M.S.M.E., 1935, Pennsylvania State University*) now superintendent of turbo-jet test projects, Test Projects Control Department, Allison Division, inventors in patent 2,775,470 for an exhaust stack vibration isolator.

Delco Appliance Division
Rochester, New York

• **Frederick Druseikis**, senior designer, Engineering Department, inventor in patent 2,762,612 for a heat exchange structure for air heating furnaces.

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

• **Raymond H. Sullivan,** senior research engineer, Engineering Department, and **Peter R. Contant,** senior project engineer, Engineering Department, inventors in patent 2,768,495 for an electric balance.

Delco Products Division
Dayton, Ohio

• **Joseph F. Bertsch,** (*B.S.M.E., University of Cincinnati, 1948*) now research engineer, Research and Development Department, Chevrolet Motor Division, **Mearick Funkhouser,** (*M.E. degree, Cornell University, 1932*) chief engineer, and **George W. Jackson,** (*B.S.M.E., Purdue University, 1937*) engineering manager, automotive and mechanical products, inventors in patent 2,761,425 for a reciprocatory fluid-actuated device.

• **Edwin F. Rossman,** (*Case Institute of Technology and B.S.M.E., Massachusetts Institute of Technology and Harvard University, 1920*) research engineer, Engineering Department, inventor in patent 2,765,054 for a self-centering cushioning device.

• **Ralph K. Shewmon,** (*General Motors Institute, 1934*) engineering manager, Research and Development Department, inventor in patent 2,766,573 for a motor with air-cooled electric motor and cutting disc.

• **John S. Wolfe,** (*B.A., The Ohio State University, 1934*) chemist, Engineering Department, inventor in patent 2,773,415 for a method and apparatus for quantitative spectral analysis.

Delco Radio Division
Kokomo, Indiana

• **James H. Guyton,** (*B.S.E.E., 1934, and M.S.E.E., 1935, Washington University*) chief engineer—radio, Engineering Department, inventor in patent 2,764,675 for a signal-actuated tuner selective over-noise level.

• **Manfred G. Wright,** (*B.S.M.E., Purdue University, 1938*) head, Mechanical Engineering Section, and **William R. Kearney,** (*B.S.M.E., Purdue University, 1933*) senior project engineer, Mechanical Engineering Section, inventors in patent 2,765,663 for an inertia drive signal seeking tuner.

• **Thomas A. Prewitt,** (*B.S.E.E., Purdue University, 1948*) senior project engineer, Electrical Laboratory, inventor in patent 2,766,384 for an automatic coil adjusting system.

Delco-Remy Division
Anderson, Indiana

• **William E. Brown,** (*General Motors Institute*) staff engineer, Product Engineering Department, and **Ward Cole,** product engineer, Engineering Department, inventors in patent 2,757,250 for direction signal mechanisms.

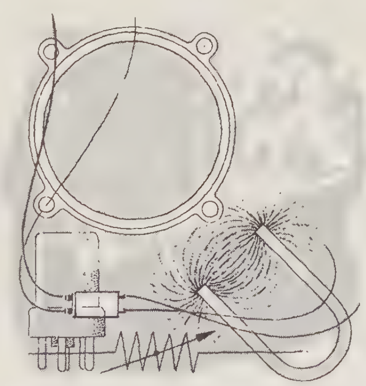
• **J. Edward Antonidis,** (*Purdue University*) section engineer, Engineering Department, inventor in patent 2,765,133 for a mounting for a dynamo electric unit.

• **Lyman A. Rice,** (*B.S.E.E., University of Utah, 1935, and M.S.E., University of Michigan, 1936*) staff engineer, Engineering Department, inventor in patent 2,766,418 for a battery charging system.

• **Richard M. Goodwin,** (*B.S.M.E., Purdue University, 1932*) senior machine designer, Process Department, inventor in patent 2,768,480 for a dynamo brush machine.

• **William E. Brown*** inventor in patent 2,768,535 for a control rod positioning mechanism.

• **Herman L. Hartzell,** (*B.E.E., The Ohio State University, 1924*) chief engineer, Engineering Department, **David C.**



Redick, (*General Motors Institute*) section head of switches and horns, Product Engineering Department, and **Clarence L. Julian,** group leader of Ignition Section, Drafting Department, inventors in patent 2,769,047 for a distributor structure.

• **Harold V. Elliott,** (*General Motors Institute*), senior project engineer, Product Engineering Department, inventor in patent 2,769,352 for a rod controlling mechanism.

• **Harold J. Cromwell,** (*B.S.M.E., Purdue University, 1933*) senior research engineer, Research Department, inventor in patent 2,773,958 for a switch.

• **J. B. Harrison,** (*B.S.Ch.E., Purdue University, 1934*) battery plant manager, and **Garth A. Rowls,** (*B.S.M.E., Purdue University, 1932*) product engineer—storage batteries, Battery Engineering Department, inventors in patent 2,774,805 for a battery structure.

• **Charles E. Buck,** (*B.S.E.E., Purdue University, 1935*) engineer—solenoids, Engineering Department, and **Argyle G. Lautzenhiser,** (*B.S.E.E., Tri-State College, 1940*) now senior project engineer, AC Spark Plug Division, inventors in patent 2,774,844 for a resettable circuit breaker.

• **J. Edward Antonidis*, William C. Edmundson,** (*B.S.E.E., Purdue University, 1934*) staff engineer, Product Engineering Department, **Clayton W. Smith,** checker and layout man, Engineering Drafting Department, and **Perry W. House,** (*B.S.Ch.E., Purdue University, 1933*) assistant chief engineer, Engineering Department, inventors in patent 2,774,894 for a connector for an electric horn.

Detroit Diesel Engine Division
Detroit, Michigan

• **John Dickson,** (*diploma, Royal Technical College, Glasgow, Scotland*) staff engineer in charge of forward design, Engineering Department, inventor in patent 2,770,500 for a fuel injection device.

• **Charles H. Frick,** (*B.S., Iowa State College, 1934*) senior project engineer, Engineering Department, and **Glenn R. Hickson,** project engineer, Experimental Test Department, GMC Truck and Coach Division, inventors in patent 2,771,788 for a control mechanism.

• **Virgin C. Reddy,** (*B.S.M.E., 1934, and M.S.M.E., 1935, Iowa State College, and General Motors Institute, 1936*) development engineer, Engineering Department, inventor in patent 2,773,491 for a pressure control valve.

Detroit Transmission Division
Ypsilanti, Michigan

• **Kenneth E. Snyder,** senior project engi-

neer, Engineering Department, inventor in patent 2,768,503 for controls for servomechanisms.

Diesel Equipment Division
Grand Rapids, Michigan

• **William J. Purchas, Jr.**, (*B.S.M.E., Detroit Institute of Technology, 1933*) now chief engineer, Bearings Department of Transmissions Operations, Allison Division, inventor in patent 2,766,745 for a hydraulic valve lifter.

Electro-Motive Division
La Grange, Illinois

• **Herbert A. Krause**, (*diploma in E.E., International School of Correspondence, 1950*) project engineer, Electrical Control Laboratory, inventor in patent 2,766,356 for an electrical switch mechanism.

• **J. Paul Miller**, (*Purdue University and Cornell University*) now on special assignment, Engineering Department, Allison Division, and **Richard M. Dilworth**, retired, inventors in patent 2,770,200 for a railway vehicle truck.

• **Torsten O. Lillquist**, electrical research engineer, Engineering Department, inventor in patent 2,773,227 for a wheel slip control system.

GM Engineering Staff
Detroit, Michigan

• **John Dolza**, (*M.S. in E.E. and M.E., Polytechnic Institute, Turin, Italy, 1926*) engineer-in-charge, Power Development Section, **George P. Ransom**, (*B.S.M.E., University of Michigan, 1949*) section engineer, Power Development Section, and **Donald C. Unger**, (*Detroit Institute of Technology and University of Michigan*) project engineer, Power Development Section, inventors in patent 2,763,252 for engines.

• **Charles A. Chayne**, (*B.S.M.E., Massachusetts Institute of Technology, 1919, and Harvard University*) vice president in charge of Engineering Staff, inventor in patent 2,765,883 for brake cooling.

• **Oliver K. Kelley**, (*B.S.M.E., Chicago Technical College, 1925, and Massachusetts Institute of Technology*) engineer-in-charge Transmission Development Section, inventor in patent 2,766,641 for a dual-range plural turbine-gear drive.

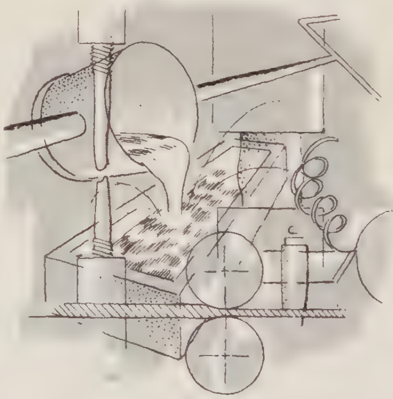
• **Bertil B. Cederleaf**, (*General Motors Institute, 1949, B.M.E., 1951*) senior engineer—experimental procurement, Purchasing Section, and **John Dolza*** inventors in patent 2,766,742 for a cylinder for an internal combustion engine.

• **John Dolza*** inventor in patent 2,768,414 for a flask and mold for a V-6 cylinder block.

• **John Dolza***, **William K. Steinhagen**, (*B.S.E., 1947, and M.S.E., 1948, University of Michigan*) assistant engineer-in-charge, Power Development Section, and **Ralph S. Johnson**, (*B.S.M.E., Purdue University, 1933*) senior project engineer, Power Development Section, inventors in patent 2,769,660 for a thermostat.

• **Von D. Polhemus**, (*B.S.M.E., University of Cincinnati, 1933*) engineer-in-charge, Structure and Suspension Development Group, and **Lothrop M. Forbush**, (*B.S., Harvard University*) staff engineer, Automotive Ordnance Section, inventors in patent 2,773,990 for a stroboscopic display device.

• **Walter W. Ridel**, (*LL.B. and Master of Patent Law degree, University of Dayton, 1926*) patent attorney, Patent Section Dayton office, inventor in patent 2,774,001 for motor stator laminations and a method of manufacturing welded stators.



Euclid Division
Cleveland, Ohio

• **Raymond Q. Armington**, (*B.I.E., The Ohio State University, 1928*) general manager, inventor in patent 2,773,319 for a conveyor loader hitch.

• **Edward R. Fryer**, (*General Motors Institute and B.S.M.E., Massachusetts Institute of Technology, 1945*) senior project engineer, Engineering Department, and **William J. Adams**, retired, inventors in patent 2,773,320 for a digging and carrying scraper.

Fabricast Division
Bedford, Indiana

• **Arthur J. Anderson**, process engineer, Process Engineering Department, inventor in patent 2,770,858 for supporting shell molds during metal pouring operations.

Fisher Body Division
Detroit, Michigan

• **Arthur T. Lausten**, (*B.S., Wayne State University, 1936*) now senior research chemist, Process Development Department, Ternstedt Division, and **Grayland T. Larsen**, no longer with the Division, inventors in patent 2,767,359 for a high voltage current control.

• **George D. Legge**, (*Western Technical Institute, Toronto, Canada*) group leader—hardware design, Trim and Hardware Styling Department, inventor in patent 2,770,388 for an ash receiver or the like.

• **Lewis J. Lamm**, (*General Motors Institute 1937. B.S.M.E., 1938, and Juris Doctor of Law degree, 1941, George Washington University; U.S. Naval Academy post-graduate school; and M.S. in communication engineering, Harvard University, 1946*) assistant plant manager, Grand Rapids plant No. 2, inventor in patent 2,773,472 for an apparatus for electrostatic spray coating.

• **Julius J. Balint**, engineer-in-charge, Engineering Department, **Gilbert A. Bannasch**, group leader, Body Engineering Department, and **David H. Bratton**, (*University of Tennessee and Art Institute of the South*) now project engineer, Passenger Body Engineering Department, Chevrolet Motor Division, inventors in patent 2,775,479 for an operating apparatus for a vehicle window.

• **Barthold F. Meyer**, (*B.S.M.E., Pratt Institute, 1939, and Johns Hopkins University*) now supervisor of mechanical design group, Product Engineering Department, Ternstedt Division, and **William F. Kapanka**, no longer with the Division, inventors in patent 2,775,783 for a door hinge, check, and hold-open.

Frigidaire Division
Dayton, Ohio

• **James W. Jacobs**, (*B.S.M.E., University of Dayton, 1954*) section engineer,

Appliance Engineering Department, and **Donald L. Coning,** no longer with the Division, inventors in patent 2,762,229 for a drive mechanism for a refrigerating mechanism.

• **Verlos G. Sharpe,** (*B.S.M.E., Purdue University, 1948*) section engineer, Household Engineering Department, inventor in patent 2,764,874 for a refrigerator door fastener.

• **George C. Pearce,** (*B.S.M.E., Stanford University, 1924*) section head, Appliance Engineering Department, and **James H. Nellis,** (*B.S.M.E., Purdue University, 1936*) senior project engineer, Appliance Engineering Department, inventors in patent 2,765,375 for a domestic appliance.

• **George B. Long,** (*B.S.E.E., Purdue University, 1937*) supervisor of major products line, Research and Future Products Engineering Department, inventor in patents 2,765,454 for clip terminals and 2,778,613 for a cooking vessel with stirrer.

• **George C. Pearce*** inventor in patents 2,765,728 for a cooking utensil including a porous metal cooking surface and 2,769,440 for a domestic appliance.

• **Alfred J. Sacksteder,** (*B.S.E.E., University of Cincinnati, 1933*) assistant superintendent, Welding Engineering Department, inventor in patent 2,766,369 for an electrical apparatus.

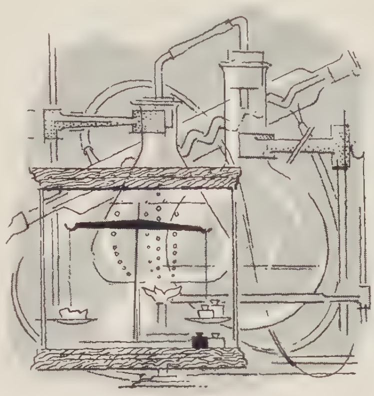
• **Keith K. Kesling,** (*University of Dayton and Dayton Art Institute*) project and design engineer, Research and Future Products Engineering Department, inventor in patents 2,767,040 and 2,767,042 for refrigerating apparatus.

• **Millard E. Fry,** (*B.S.M.E., University of Pittsburgh, 1931*) senior project engineer, Engineering Department, inventor in patent 2,767,298 for a domestic appliance.

• **Marshall C. Harrold,** (*B.S.M.E., Purdue University, 1931*) senior project engineer, Engineering Department, inventor in patent 2,769,048 for an electrical apparatus.

*Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.

• **Leland H. Grenell,** (*B.S., Pennsylvania State College, 1924*) supervisor, Metals Section, inventor in patents 2,769,231 for a method of joining an aluminum tube to a steel refrigerator compressor and 2,769,318 for a refrigerating apparatus of dissimilar metals.



• **Marshall C. Harrold* and Harold A. Wheeler,** no longer with the Division, inventors in patent 2,769,312 for a refrigerant expansion control.

• **George H. Strickland,** (*Harvard University and LL.B., George Washington University, 1929*) assistant director, Patent Section Dayton office, inventor in patent 2,769,317 for a freezing device.

• **William Smith, Jr.,** (*B.S. in chemistry and physics, Pennsylvania State University, 1927*) section head, Appliance Engineering Department, inventor in patent 2,769,327 for a domestic appliance.

• **Richard S. Gaugler,** (*B.S.Ch.E., Purdue University, 1922*) supervisor of major product line, and **Robert Galin,** (*B.S.M.E., Robert College, Istanbul, Turkey, 1947, and M.S.M.E., University of Michigan, 1949*) senior project engineer, Research and Future Products Engineering Department, inventors in patent 2,772,542 for an ice tray.

• **James A. Canter,** (*B.M.E., The Ohio State University, 1936*) senior project engineer, Commercial Engineering Department, inventor in patent 2,773,357 for an open top display case with night cover.

• **Marshall W. Baker,** (*B.M.E., The Ohio State University, 1925*) manager, Commercial and Air Conditioning Engineering Department, **Hal C. Johnston,** (*Wayne State University and Lawrence Institute of Technology*) supervisor of major product line, Commercial Engineering Department, and **Charles F. Henney,** retired, inventors in patents 2,773,360 for a vehicle refrigerating apparatus and 2,779,162 for an automobile refrigerating apparatus.

• **James W. Jacobs* and Clifford H. Wurtz,** (*B.S., University of Illinois, 1929*) supervisor of major product line, inventors in patent 2,773,361 for a refrigerator cabinet and evaporator structure.

• **Curtis P. Kelley,** (*B.S.M.E., University of Kentucky, 1936*) senior project engineer, Appliance Engineering Department, inventor in patent 2,774,219 for an automobile refrigerating apparatus.

• **John H. Heidorn,** (*General Motors Institute*) section engineer, Engineering Department, inventor in patent 2,774,221 for a two-temperature refrigerating system.

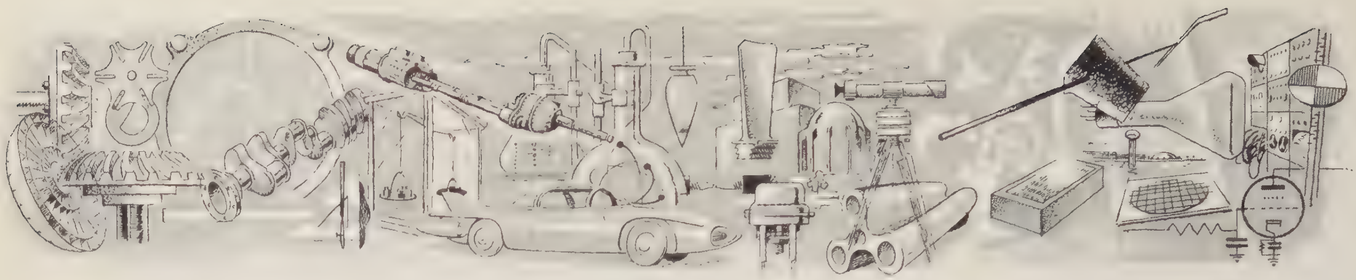
• **James W. Jacobs* and Daniel J. Barbulesco,** (*B.S.M.E., Stanford University, 1949*) project engineer, Appliance Engineering Department, inventors in patent 2,774,222 for a vehicle refrigerating apparatus.

• **Edmund F. Schweller,** assistant chief engineer, inventor in patents 2,774,226 and 2,778,201 for freezing devices.

• **Edward C. Simmons,** (*University of Dayton*) senior engineer, Household Engineering Department, inventor in patent 2,775,102 for a freezing device.

GM Overseas Operations Division
New York, New York

• **Harry R. Stocks,** assistant chief engineer, Vauxhall Motors, Ltd., Luton, England, **Thomas C.F. Stott,** (*M.I.M.E., EastHam Technical College, England, 1925*) passenger vehicle engineer, Engineering Department, Vauxhall Motors, Ltd., Luton, England, and **Frank Wescott,** Special Assignments, Engineering Department, Vauxhall Motors, Ltd., Luton, England, inventors in patent 2,773,399 for a steering column transmission control mechanism.



• **Alfred Doerper,** (*engineering degree, Technical University, Aachen, Germany, 1924*) superintendent of experimental shop, Product Engineering Department, Adam Opel, A.G., Russelsheim/Main, Germany, inventor in patent 2,775,136 for a steering gear for motor vehicles.

• **Waldemar Assmus,** (*engineering degree, Polytechnical Institute, Mittweida, Germany, 1923*) truck engineer, Engineering Department, Adam Opel, A.G., Russelsheim/Main, Germany, inventor in patent 2,776,135 for springing of road vehicles.

GMC Truck & Coach Division
Pontiac, Michigan

• **Hans O. Schjolin,** (*Karlstad College, Sweden, 1920, and Polytechnical Institute, Mittweida, Germany, 1923*) new development engineer, Engineering Department, and **Donald K. Isbell,** senior engineer, Engineering Department, inventors in patents 2,765,052 for wear adjustment for brakes and 2,773,552 for a drive wheel and brake assembly.

• **Hans O. Schjolin* and Karl Schuster,** (*B.S., Michigan State University, 1928*) project engineer, New Development Department, inventors in patent 2,774,227 for an air conditioning apparatus for buses and the like.

• **Hans O. Schjolin*, Helmuth Guentsche,** (*engineering degree, Technical University of Berlin, Charlottenburg, Germany, 1923*) drafting supervisor, Engineering Department, and **Millis V. Parshall,** (*B.S.M.E., University of Michigan, 1918*) project engineer, New Development Department, inventors in patent 2,775,330 for a transmission and control system.

Guide Lamp Division
Anderson, Indiana

• **George W. Onksen,** (*General Motors Institute, 1933, B.I.E., 1956*) Engineering Head, Research and Development, **John W. Cole,** Designer, Engineering Department, and **Raymond A. Gaither,** no longer with the Division, inventors in patent 2,762,912 for a lens.

• **Robert N. Falge,** (*B.S.E.E., University of Wisconsin, 1916*) technical assistant to the general manager, **George W. Onksen*, Harold Todd,** (*B.S.E.E., Purdue University, 1940*) senior project engineer—factory and field service problems, Engineering Department, and **Charles W. Miller,** (*Purdue University*) project engineer, Engineering Department, inventors in patent 2,762,932 for a light pickup unit.

• **George W. Onksen* and Carl A. Hokans,** no longer with the Division, inventors in patent 2,767,305 for a vehicle lamp.

• **George W. Onksen* and Raymond A. Gaither*** inventors in patent 2,767,306 for a composite beam vehicle headlamp.

• **Charles W. Miller*, Harold Todd*,** and **George W. Onksen*** inventors in patent 2,767,347 for an automatic headlight dimmer system.

• **Robert N. Falge*** inventor in patent 2,769,157 for a molded electrical connection.

• **Howard C. Mead,** (*Western Reserve University*) chief engineer, inventor in patent 2,770,714 for a plastic shield for a flush-mounted headlamp.

• **James E. Kingery,** foreman, Inspection Department, inventor in patent 2,770,716 for a T-shaped split beam tractor lamp.

• **Charles W. Miller* and Harold Todd*** inventors in patent 2,771,569 for an automatic headlight dimmer system—high voltage switching.

Harrison Radiator Division
Lockport, New York

• **Rodney M. Johnston,** senior engineer, Engineering Department, inventor in patent 2,778,576 for temperature responsive actuating devices.

• **William H. Lloyd,** senior designer, Engineering Department, **Frank A. Disinger,** product designer, Engineering Department, and **J. Robert Hayden,** supervisor—design, Engineering Department, inventors in patent 2,778,606 for heat exchangers.

Inland Manufacturing Division
Dayton, Ohio

• **Max P. Baker,** (*A.B., Miami University, 1922*) project engineer, Engineering Department, inventor in patent 2,761,296 for a drive shaft.

• **Paul E. Clingman,** (*General Motors Institute, 1935*) supervisor, Quality and Control Department, and **G. William Beck,** (*B.I.E., General Motors Institute, 1947*) assistant chief engineer—product design, Engineering Department, inventors in patent 2,763,345 for a connector strip.

• **Paul E. Clingman*** inventor in patent 2,768,411 for sealing strips.

Moraine Products Division
Dayton, Ohio

• **Earl W. Reinsch,** (*B.S.Met.E., Michigan College of Mining and Technology, 1940*) Supervisor, Experimental Test Department, Engineering Department, inventor in patent 2,768,748 for a separator.

New Departure Division
Bristol, Connecticut

• **Leland D. Cobb,** manager, Research and Development Laboratory, inventor in patent 2,764,443 for a demountable closure.

Oldsmobile Division
Lansing, Michigan

• **Gilbert Burrell,** (*B.S.E.E., Michigan State University, 1928*) motor engineer, Product Engineering Department, inventor in patent 2,771,064 for crankcase ventilation.

Pontiac Motor Division
Pontiac, Michigan

• **Lester V. Ostrander,** assistant chassis engineer, Engineering Department, inventor in patent 2,765,780 for an intake and exhaust manifold.

• **Mark H. Frank,** (*B.S.M.E., Michigan State University, 1927*) motor engineer, Engineering Department, inventor in patent 2,766,634 for a timing chain bumper.

• **Stuart R. Kern,** (*B.S.M.E., University of Michigan, 1929*) carburetor engineer, Engineering Department, inventor in patent 2,766,748 for an intake manifold.

• **Clayton B. Leach,** (*A.B. in mathematics and chemistry, Park College, 1934, and General Motors Institute*) chassis engineer, Engineering Department, inventor in patent 2,771,869 for a gasket.

• **Robert R. Hutchison,** (*B.S.M.E., Purdue University, 1913*) now engineering consultant, Adam Opel, A.G., Russelsheim Main, Germany, inventor in patent 2,771,870 for a gasket.

GM Research Staff
Detroit, Michigan

• **Alfred W. Schluchter,** (*B.S. in chemistry, 1919, M.S. in chemistry, 1921, and Ph.D. in physical chemistry, 1926, University of Michigan*) research engineer, Mechanical Development Department, inventor in patents 2,766,116 for an aluminum base bearing alloy and 2,770,031 for a bearing.

• **F. E. Heffner,** (*B.S.M.E., 1948, and M.S.M.E., 1950, Wayne State University*) senior research engineer, Mechanical Development Department, inventor in patent 2,766,624 for a test chamber.

• **Edward J. Martin,** (*B.S., 1915, M.S. in physics, 1917, and Ph.D. in physics, 1924, University of Michigan*) head, Physics and Instrumentation Department, and **Robert N. Frawley,** deceased, inventors in patent 2,766,653 for a radiant energy transmission and reflector analyzer with adjustable filter.

• **Robert Schilling,** (*M.E. degree, Technical University, Munich, Germany, 1922*) now director of research and development, Chevrolet Motor Division, and **Frank J. Rowlandt,** no longer with the Staff, inventors in patent 2,766,864 for a fluid-operated torque-responsive clutch device.

• **Donald J. Henry,** (*M.S., The Ohio State University, 1937*) assistant head, Metallurgical Engineering Department, inventor in patents 2,770,859 for a mold release agent for shell molding and 2,772,458 for a method of making smooth-surfaced sand-resin molds.

• **Fred J. Webbere,** (*B.S., University of Wisconsin, 1941*) supervisor of alloy-development melting, Metallurgical Engineering Department, inventor in patents 2,770,860 for casting readily oxidizable alloys and 2,772,457 for a method of shell molding.

• **Darl F. Caris,** (*B.S.E.E., 1926, and professional degree of E.E., 1932, University of Michigan*) head, Automotive Engines Department, and **Fred Davis,** retired, inventors in patent 2,771,864 for a ring type manifold.

• **William F. King,** (*General Motors Institute, 1940 and B.S.M.E., University of Michigan, 1941*) assistant head, Special Problems Department, inventor in patent 2,773,389 for a balancing machine indicator.

Rochester Products Division
Rochester, New York

• **Joseph M. McDonnell,** senior engineer, Engineering Department, and **Henry D. Mowers,** production engineer, Production Engineering Department, inventors in patent 2,763,340 for an antenna mechanism.

• **Elmer Olson,** director of sales and engineering, and **Lawrence C. Dermond,** (*Purdue University and Tri-State College*) senior research engineer, Engineering Department, inventors in patent 2,771,282 for a carburetor.

• **Richard W. Spears,** (*B.S.M.E., University of Rochester, 1941*) superintendent of manufacturing—Tube Plant, Factory Management Department, **Richard D. Williams,** (*University of Rochester, Columbia University, and Rochester Institute of Technology*) project engineer, Product Engineering and Development Department and **John W. Armstrong,** no longer with the Division, inventors in patent 2,771,669 for a method of coating the interior of tubing with zinc.

Saginaw Steering Gear Division
Saginaw, Michigan

• **C. W. Lincoln,** (*B.S.M.E., University of Illinois, 1916*) chief engineer, and **Philip B. Zeigler,** (*B.S.E., Purdue University, 1941*) assistant chief engineer, Product Engineering Department, inventors in patent 2,768,531 for a power steering apparatus.

• **C. W. Lincoln* and Henry D. Spiekerman** and **Robert M. Gold,** no longer with the Division, inventors in patent 2,775,664 for a neutral safety and back-up light switch.

GM Styling Staff
Detroit, Michigan

• **Louis P. Garvey,** (*B.M.E., University of Detroit, 1940*) senior project engineer, Experimental and Development Department, Fisher Body Division, **Karl A. Walter,** (*University of Michigan*) group leader, Experimental and Development Department, Fisher Body Division, and **Robert L. Ballard,** (*B.S.M.E., 1948, and B.S.E.E., 1949, Worcester Polytechnic Institute*) senior process engineer-in-charge of mechanization development, Process Development Department, Harrison Radiator Division, inventors in patent 2,770,489 for a vehicle folding top structure and power actuating means therefor.

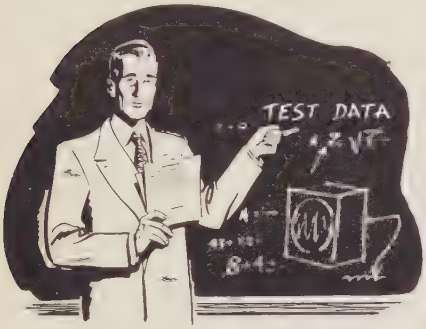
• **Harley J. Earl,** (*Stanford University*) vice president in charge of Styling Staff, inventor in patent 2,771,542 for a headlamp and door assembly.

• **Robert McClure,** (*Rolls Royce, Ltd., 5-year engineering program*) senior project engineer, Special Studio No. 1, and **Joseph M. Stimetz,** deceased, inventors in patent 2,775,478 for a vehicle door hinge and cam.

These patent listings are informative only and are not intended to define the coverage which is determined by the claims of each one.

Technical Presentations by GM Engineers

The technical presentation is another way in which information about current engineering and scientific developments in General Motors can be made available to the public. A listing of speaking appearances by General Motors engineers, such as that given below, usually includes the presentation of papers before professional societies, lecturing to college engineering classes or student societies, and speaking to civic or governmental organizations. Educators who wish assistance in obtaining the services of GM engineers to speak to student groups may write to the Educational Relations Section, Public Relations Staff, General Motors Technical Center, P. O. Box 177, North End Station, Detroit 2, Michigan.



GM personnel who have made recent presentations are as follows:

Automotive Engineering

Rudolph J. Gorsky, staff engineer, Transmission Division, Engineering Department, Buick Motor Division, before the Erie Chapters of the A.S.M.E. and the Pennsylvania Society of Professional Engineers, Erie, Pennsylvania, January 28; title: The Dynaflow Transmission.

M. M. Roensch, director of laboratory tests, Experimental Testing Department, Chevrolet Motor Division, before the S.A.E., Ponca City, Oklahoma, January 25; title: Chevrolet Fuel Injection System Development.

S. C. Richey, staff engineer, Electrical and Accessory Department, Chevrolet Motor Division, before the Cleveland Section Ladies' Unit of the S.A.E., Cleveland, Ohio, February 11; title: Carriage for Milady.

Wayne A. Bonvallet, project engineer, Noise and Vibration Laboratory, GM Proving Ground, before the Detroit Section of the Society of Automotive Engineers, Detroit, February 18; title: The Ins and Outs of Motor Truck Noise.

C. F. Nixon, head, Electrochemistry Department, GM Research Staff, before the S.A.E., Syracuse, New York, February 18; title: The Bright Look.

J. B. Burnell, assistant staff engineer, Engine Design Department, Chevrolet Motor Division, before the Society of Fleet Supervisors, New York City, February 20; and before the S.A.E. Mid-Michigan Section, Flint, Michigan, April 8; title: The Chevrolet Ramjet Fuel Injection System.

N. E. Farley, director, Engineering Laboratory, Chevrolet Motor Division, before the Engineers Club of Virginia Peninsula, Newport News, Virginia, February 21; title: Why Horsepower?

Robert K. Hathaway, sales and service engineer, Product Sales and Service Department, **Warren F. Williams,** senior project engineer, Product Engineering and Development Department, and **Elmer F. DeTiere,** senior project engineer, Research and Development Department, Rochester Products Division, before the Genessee Valley Timing Association, Rochester, New York, February 25; title: Fuel Injection, Carburetion, and Multiple Carburetors.

James F. Hughes, director of public relations, and **Robert K. Hathaway,** sales and service engineer, Product Sales and Service Department, Rochester Products Division, over Station WHEC-TV, Rochester, New York, February 25; title: Rochester Products Division and Fuel Injection.

Donald Stoltman, senior project engineer, Research and Development Department, Rochester Products Division, before students and faculty of Edison Technical and Industrial High School, Rochester, New York, February 28; title: Fuel Injection.

G. J. Englehard, assistant staff engineer, Passenger Car Advanced Body Design Department, Chevrolet Motor Division, before S.A.E. annual meeting January 14; and before the American Society of Body Engineers, Detroit, February 28; title: Quality Car Structure.

H. K. Gandelot, engineer-in-charge, Vehicle Safety Development Group, GM Engineering Staff, before the Central Michigan Driver Training Instructors, Flint, Michigan, February 23; title: Safety Advancement in Automobiles; before the women's division of the Greater Lansing Safety Council, Lansing, Michigan, February 28; title: Designing Automobiles for Safety; and before the annual meeting of the Harvey Cushing Society in Detroit, April 26; title: Various Techniques Employed in the Crash Testing of Automobiles.

Kenneth F. Lingg, service manager, Product Sales and Service Department, Rochester Products Division, before the Automotive Electric Association, San Francisco, February 22; Seattle, Washington, March 1; Columbus, Ohio, March 9; Philadelphia, March 15; and Atlanta, Georgia, March 26; and before the Canadian Automotive Electric Association, Montreal, Canada, March 13; title: Fuel Injection.

V. D. Polhemus, engineer-in-charge, and **L. J. Kehoe,** administrative engineer, Structure and Suspension Development Group, GM Engineering Staff, before the S.A.E. national passenger car body and materials meeting, Detroit, March 7; title: Cadillac's Air Suspension for the Eldorado Brougham: Experimental Development of the Air Spring.

Rudolph J. Gorsky, staff engineer, Engineering Department, Buick Motor Division, before the Lynite Club of Aluminum Company of America, Detroit, March 11; title: Dynaflow Transmission.

Leonard C. Rowe, senior research chemist, Chemistry Department, GM Research Staff, before the National Association of Corrosion Engineers, St. Louis, March 12; title: An Evaluation of Inhibitors for Corrosion Prevention in an Engine Cooling System.

George Brown, contact engineer, Spark Plug Engineering Department, AC Spark Plug Division, before the motor vehicle maintenance conference, Seattle, Wash-

ington, March 25; title: Spark Plugs and Electrical Components; and March 26; title: Shop Procedures; before service personnel of Kroger Company, Kansas City, Missouri, April 3, and Pittsburgh, April 18; title: Spark Plugs and Shop Procedures; and before personnel of the City of Pittsburgh, April 19; title: Spark Plugs and Shop Procedures.

C. V. Crockett, chief engineer, Engineering Department, GMC Truck and Coach Division, before the ninth annual Harry E. Salzberg memorial lecture and transportation conference, Syracuse, New York, April 2; discussion of paper: Trends in Truck and Tractor Design, by Carl A. Lindblom of International Harvester Company.

R. E. Kaufman, assistant staff engineer, Truck Transmission Department, Chevrolet Motor Division, before the Michigan Trucking Association, Detroit, April 2; title: The Chevrolet Powermatic Transmission.

Robert K. Hathaway, sales and service engineer, Product Sales and Service Department, Rochester Products Division, before the Finger Lakes Region Sports Car Club of America, Rochester, New York, April 3; title: Fuel Injection.

Thomas E. Dolan, experimental engineer, Engineering Department, Detroit Transmission Division, before the General Motors Institute Technical Club, Plymouth, Michigan, April 3; title: Problems Detroit Transmission Division Had to Face in '57-'58 with Engineering Changes and How They Were Solved.

Robert F. Thomson, head, Metallurgical Engineering Department, GM Research Staff, before the American Society for Metals, Hartford, Connecticut, April 9; title: The Automotive Gas Turbine.

Arthur F. Underwood, manager of Research Staff Activities, GM Research Staff, before the S.A.E., New York City, April 11; title: Free Piston Engines Are Here.

John G. Locklin, project engineer, Structures and Suspension Department, GMC Truck and Coach Division, before students of General Motors Institute, Pontiac, Michigan, April 17; title: Why Air Suspension?

Zora Arkus-Duntov, research development engineer, Special Design and Development Department, Chevrolet Motor Division, before combined engineering groups of Princeton University, Princeton, New Jersey, April 17; title: The Chevrolet Ramjet Fuel Injection System.

F. G. Torrance, staff engineer, Production Engineering Department, Chevrolet Motor Division, before the Rotary Club, Traverse City, Michigan, April 23; title: Why Horsepower?

James J. Hovorka, project engineer, Brake Engineering Department, Moraine Products Division, before the S.A.E., Purdue University, Lafayette, Indiana, May 2; title: Crash Research in Vehicle Safety.

J. E. Eshbaugh, staff engineer, Automotive Engineering Department, AC Spark Plug Division, before the National Automotive Radiator Service Association, St. Louis, May 4; title: AC Radiator Pressure Caps.

Robert R. Mandy, supervising engineer, Automotive Air Conditioning Department, Harrison Radiator Division, before the Boston, Massachusetts Section of the American Society of Refrigerating Engineers, May 7; title: Future Trends in Automotive Air Conditioning.

Robert O. Ellerby, assistant electrical engineer, Electrical Department, GMC Truck and Coach Division, before the American Trucking Association forum, New York City, May 9; panel member on Two-Way Radios on Trucks.

F. J. Winchell, staff engineer and **W. D. Route**, assistant staff engineer, Passenger Car Transmission Department, Chevrolet Motor Division and **O. K. Kelley**, engineer-in-charge, Transmission Development Section, GM Engineering Staff, before the S.A.E. annual meeting, Detroit, January 16; title: The Chevrolet Turboglide Transmission.

Bearings and Lubrication

Roy L. Bowers, staff engineer, Automotive Engineering Department, AC Spark Plug Division, before the Western Railroad Truck Line Association, Kansas City, Missouri, February 5; title: Filtration of Automotive Crankcase Oils.

Virgil C. Hutton, lubrication engineer, Central Supplies Department, Delco Products Division, before the American Society of Lubricating Engineers, Columbus, Ohio, February 28; title: Functional Evaluation of Hydraulic Oils.

C. R. Gillette, chief chemist, Product Engineering—Chemical Section, New Departure Division, before the Connecticut Section A.S.L.E., New Haven, Connecticut, February 27; title: Ball Bearing Lubricants; and before the American Society of Mechanical Engineers, New York City, March 28; title: Ball Bearing Lubricants.

L. D. Cobb, manager, research and development, Product Engineering Department, New Departure Division, before the S.A.E. National Passenger Car Body and Material Meeting, Detroit, March 6; title: The Effect of Vacuum Melting on Bearing Steel; and before the Aircraft Ball Bearing Symposium, Hartford, Connecticut, April 17; title: Ball Bearing Research for Jet Engine Bearings.

Heinz Hanau, supervisor, aircraft projects, Product Engineering—Aircraft Section, New Departure Division, before the A.S.M.E., Cleveland, April 4; title: Turbo Accessory Bearings, and before the Aircraft Ball Bearing Symposium, Hartford, Connecticut, April 17; title: Recent Developments—Turbo Accessory Bearings.

T. W. Selby, senior research engineer, Fuels and Lubricants Department, GM Research Staff, before the American Chemical Society spring session, Miami, April 8; title: The Low-Temperature Viscometry of Oils Containing Polyisobutylene and Polyalkylmethacrylate as Viscosity; and before the A.S.L.E. national convention, Detroit, April 15; title: The Non-Newtonian Characteristics of Lubricating Oils.

Robert H. Guy, lubrication engineer, Master Mechanics Department, Detroit Transmission Division, before the A.S. L.E. national convention, Detroit, April 16; title: Automation Lubrication Practices in the Automotive Industry.

Diesel Engines

F. S. Driscoll, marine application engineer, Marine Application Section, Detroit Diesel Engine Division, before the S.A.E., Boston, March 5; title: Application of Diesel Engines to the Marine Industry.

W. E. Whitmer, project engineer, Engine Department, GMC Truck and Coach Division, before the S.A.E., Kansas City, Missouri, March 28; title: Tailoring Diesel Engines and Accessories for More Economy.

W. K. Simpson, technical director of fuels and lubricants, Engineering Department, Electro-Motive Division, before the A.S.L.E. national convention, Detroit, April 17: title; Filtration of Diesel Engine Oils.

Electrical Engineering

Rufe Hogard, sales engineer, Fort Worth, Texas, Sales Department, Delco Products Division, before a Farm Equipment Meeting, Irving, Texas, December 5; title: Application of Motors on the Farm and in the Oil Fields.

Brooks Short, supervisor of engineering research, Engineering Department, Delco-Remy Division, before the Rotary Club, Muncie, Indiana, February 5; title: Ham Radio.

Lester R. Tansek, sales engineer, Sales Department, Delco Products Division, before the Refrigeration Service Engineers Society, New Haven, Connecticut, February 6; title: Refrigeration Motors—Selection and Maintenance.

T. N. Schierloh, service manager, Service Department, Delco Products Division, before the National Industrial Service Association, Chicago, February 12; title: Single-Phase Motors—Past, Present, and Future; and before the Refrigeration Service Engineers Society, Michigan state convention, Saginaw, April 27; title: Refrigeration Motor Servicing.

R. K. Shewmon, manager, Research and Development Engineering Department, Delco Products Division, before the student branch of the American Institute of Electrical Engineers, Tennessee Polytechnic Institute, Cookeville, Tennessee, February 15; title: Motor Design—Application Coordination.

Donald O. Ruff, senior project engineer, Electrical Engineering Department, Delco Radio Division, before the Chicago Acoustical and Audio Group, Chicago, February 20; title: High Power Transistors for Audio Circuitry.

George B. Hardenbrook, section head, Product Engineering Department, and **Richard Bartek,** project engineer, Electronic Engineering Development Department, AC Spark Plug Division, before the Saginaw Valley Chapter of the A.I.E.E., Saginaw, Michigan, March 25; title: Digital Computers.

F. E. Jaumot, Jr., director of semiconductors research and engineering, Semiconductors Department, Delco Radio Division, before the winter quarter commencement ceremonies of the Milwaukee School of Engineering, Milwaukee, Wisconsin, March 25; title: The Future of the Electrical Engineer and Engineering Technician in the Electronics and Communications Industries.

W. E. Menzies, section engineer on d-c motors and generators, Engineering Department, Delco Products Division, before the A.I.E.E., Dayton, Ohio, April 17; title: A Balanced Network Multi-Circuit Direct Current Armature Winding with No Equalizing Currents.

General Engineering

D. F. Eary, instructor, Industrial Engineering Department, General Motors Institute, before a student group at Michigan State University, East Lansing, Michigan, January 15; title: Employment of Statistical Methods of Evaluating Test Data.

C. R. Russell, assistant head, Nuclear Power Department, GM Research Staff, before the S.A.E. annual meeting, Detroit, Michigan, January 18; title: The Safety Aspects of the Use of Nuclear Energy.

James A. Norton, senior experimental chemist, Research Department, AC Spark Plug Division, before the Saginaw Valley Section of the A.S.L.E., Saginaw, Michigan, January 31; title: Wyoming's Pressure Cooker Country with accompanying film.

Kirk Reid, test engineer, Experimental Testing Department, Euclid Division, before the Society of Experimental Stress Analysis, Cleveland, Ohio, January 28; title: Laboratory and Field Testing of Euclid Earthmoving Scrapers.

George W. Beck and **Robert E. Authiel,** assistant chief engineers, Engineering Department, Inland Manufacturing Division, before the Southern Ohio Rubber Group, Dayton, February 20; titles: Extruded Rubber Sections and Automotive Hydraulic Hose, respectively.

J. H. Anderson, project engineer, Automotive Engineering Department, AC Spark Plug Division, before the Men's Club of the Presbyterian Church, Fenton, Michigan, February 22; title: What is Creative Thinking? and before the Lake Fenton Kiwanis Club, Fenton, Michigan, March 27; title: Creative Thinking.

C. E. Fausel, superintendent, Maintenance Department, Central Foundry Division, before a group of engineering students, University of Illinois, Chicago, February 22; title: Opportunities of the Foundry.

Olin A. Lee, field engineer, Sales Development Department, Euclid Division, before a group of forestry students, University of Washington, Seattle, Washington; title: Application of Log Hauling Equipment and Road Construction Machinery for Building Logging Roads.

Wallace E. Wilson, general manager, Rochester Products Division, before the Rochester Section of the Controllers Institute of America, Rochester, New York, February 25; title: Good Soldiers—Why? A Discussion of Management Philosophy.

Kenneth A. Meade, director, Educational Relations Section, GM Public Relations Staff, before the student chapter of the Society for the Advancement of Management, University of Houston, Houston, Texas, March 6; title: Some Management Challenges; before the National Science Teachers Association, Cleveland, March 21; title: Science Education Today for Tomorrow; before engineering freshmen, University of Michigan, Ann Arbor, Michigan, May 1; title: What is Engineering?; and before students of Wilbur Wright High School, Detroit, May 9; title: Career Opportunities Through Cooperative Education.

Robert K. Burns, director of education, Personnel Relations Department, Delco Products Division, before students of Sinclair College, Dayton, Ohio, March 1; title: Opportunity and Challenge in Cooperative Education.

Before airline personnel and distributors, Flint, Michigan, March 11 and 14: **W. C. Cole, H. E. Fortier,** and **W. R. Houser,** senior project engineers, Spark Plug Engineering Department, AC Spark Plug Division; title: Aircraft Spark Plugs; and **Alfred Candelise,** staff engineer, and **H. H. Vogel,** spark plug engineer, Spark Plug Engineering Department, AC Spark Plug Division; title: Aircraft Spark Plugs.

Leonard E. A. Batz, section head, Design and Standards Department, AC Spark Plug Division, before the Flint Shrine Club, Flint, Michigan, March 5, and before the East Flint Optimist Club, Flint, Michigan, May 9; title: Rockets or Missiles; before the fundamentals of engineering design class, General Motors Institute, Flint, Michigan, March 12, and before the Saginaw Kiwanis Club, Saginaw, Michigan, April 25; title: Engineering Law and Ethics.

Jack E. Feldman, technical superintendent, Quality and Control Department, Inland Manufacturing Division, before the Southern Ohio Rubber Group,

Dayton, March 20; title: Chemical Blown Sponge.

Alfred Candelise, staff engineer, Spark Plug Engineering Department, AC Spark Plug Division, before Air France personnel, Paris, France, March 22; title: Aircraft Spark Plugs.

John D. Coleman, staff engineer, Chief Engineer's Office, Frigidaire Division, before the Engineer in Industry Functional Group at the annual convention of the Ohio Society of Professional Engineers, Columbus, March 23; title: What Industry Expects of Its Engineering Employes.

William A. Frye, chief test engineer, Engineering Department, Inland Manufacturing Division, before the Southern Ohio Rubber Group, Dayton, April 3; title: Rubber Specifications in Testing.

Harvey R. Tuck, senior project engineer, Research and Future Product Engineering Department, Frigidaire Division, before the GM Engineering Executive Class in Computer Applications, I.B.M. Corporation, Poughkeepsie, New York, April 11; title: Engineering Applications of Electronic Computers at Frigidaire.

R. W. Leland, manager, Electrical Products Engineering Department, Delco Products Division, before the A.I.E.E., Dayton, Ohio, April 10; panel member: Research Approach to Engineering Problems.

F. H. McCormick, assistant chief engineer, Appliance Engineering Department, Frigidaire Division, before the American Home Laundry Manufacturers Association, French Lick, Indiana, April 16; chairman of forum on Future Designs of Home Laundry Equipment.

Martin J. Caserio, director of engineering equipment sales, AC Spark Plug Division, before the National Association of Purchasing Agents, Warehouse Distributors, Boca Raton, Florida, April 29; title: Effect of Engineering Design on Replacement; and before the National Institute of Ceramic Engineers, Dallas, Texas, May 5; title: Parts Business and Creativity.

Nathan Weingarden, chief chassis draftsman, Chassis Drafting Department, Pontiac Motor Division, before a group of students, Pontiac Senior High School, Pontiac, Michigan, April 30; title: Drafting—A Gilt-Edged Vocation.

O. G. Saettel, sales engineer, Sales Department, Delco Products Division, before the Sales Engineering Division of the Cleveland Engineering Society, Cleveland, May 3; title: Prospecting.

J. A. McDougal, assistant chief engineer, Automotive Engineering Department, AC Spark Plug Division, before the Michigan physics teachers convention, Flint, May 10; member of panel discussion on physics teaching.

Industrial Engineering

William I. Snover, manager, Production Engineering Department, Brown-Lipe-Chapin Division, before the American Institute of Industrial Engineers, Syracuse, New York, February 12; title: Cost Control Pre-planning for 1-year Model Runs.

Edward R. Clark, supervisor of quality control, Inspection Department, Detroit Transmission Division, before the Quality Control Forum, Windsor, Ontario, February 16; title: Vendors—Vendees and Quality.

John S. Marratt, plant manager, Hyatt Bearings Division, before the Newark Youth Council, Newark, New Jersey, February 24; discussion leader: Youth Wants to Know About Automation.

Russell F. Holmes, standards engineer, Automotive Engineering Department, AC Spark Plug Division, before the Standards Engineering Society of Detroit, February 25; title: The AC Engineering Standards Department.

Arnold Lichner, project engineer, Appliance Engineering Department, Frigidaire Division, before the Dayton Chapter of the American Society for Quality Control, Dayton, Ohio, March 2; conference chairman for the southern Ohio conference.

John Q. Holmes, director, Production Engineering Section, GM Process Development Staff, before the silver anniversary convention of the American Society of Tool Engineers, Houston, Texas, March 27; title: Conception and Development of JIC Standards.

Arthur Bender, quality control engineer, Engineering Department, Delco-Remy Division, before the American Society for Quality Control, Hartford, Connecticut, March 30; title: Probability Paper.

William Mertens, chief metallurgist, Master Mechanics Department, Detroit Transmission Division, before the American Welding Society, Philadelphia, April 9; title: Inert Gas, Tungsten Arc Spot Welding of an Automotive Transmission Sub-Assembly.

Donald H. Johnson, supervisor of methods engineering, Work Standards and Methods Engineering Section, GM Process Development Staff, before the student chapter of the Society for Advancement of Management, University of Detroit, April 26; title: Methods Engineering in General Motors.

R. D. McLandress, director, Work Standards and Methods Engineering Section, GM Process Development Staff, before the Detroit Chapter of the Society for Advancement of Management, May 2; panel moderator on Controlling Indirect Labor.

Leo J. Nartker, superintendent, Quality Control Department, Delco Products Division, before the Society for the Advancement of Management, Columbus, Ohio, May 2; title: Quality Control Techniques.

Manufacturing

Kenneth D. Musser, senior process engineer, Guide Lamp Division, before the student chapter of the A.S.M.E., University of Missouri, Columbia, Missouri, February 11; title: The Process Engineer's Role in Industry.

R. C. Robinson, superintendent, Standards and Methods Department, Central Foundry Division, before the American Foundrymens Society, Rolla, Missouri, February 6; title: The Manufacture of ArmaSteel Crankshafts.

R. B. Colten, staff engineer, Electronics Department, GM Process Development Staff, before the Saginaw Valley Instrumentation Engineers, Flint, Michigan, February 27; title: Ultrasonics and Other Methods of Non-Destructive Testing; before the machine tool electrification forum, Buffalo, New York, April 25; title: Standards for Industrial Equipment; and before the American Society of Safety Engineers, Warren, Ohio, May 13; title: Ultrasonics Safety Inspection.

R. R. Farley, senior project engineer, Engineering Department, GM Process Development Staff, before the S.A.E., Buffalo, New York, March 20; title: Integration of Inspection and Automation.

Glen R. Fitzgerald, chief engineer, Automotive Engineering Department, AC Spark Plug Division, before the S.A.E. national production meeting, Buffalo, New York, March 21; title: Process Improvements Being Developed through Manufacturing Research.

J. C. Holzwarth, supervisor, Metallurgical Engineering Department, GM Research Staff, before the S.A.E. national production meeting, Buffalo, New York, March 21; title: Wear-Resistant, Low-Cost Sheet Metal Forming Dies.

L. J. Pedicini, staff engineer, Foundry Department, GM Process Development Staff, before the Central Michigan Chapter of the American Foundrymens Society, Battle Creek, Michigan, May 2; title: Change Molding Methods and Why.

H. F. Barr, chief engineer, Chevrolet Motor Division, before the American Foundrymens Society, Cincinnati, Ohio, May 6; title: An Automotive Engineer Views the Foundry.

Walter E. Mitchell, general administrator, Paint Standards, Manufacturing Staff, Fisher Body Division, before the National Paint, Varnish, and Lacquer Association Technical Committee, New York City, May 15; title: Problems and Difficulties Encountered In Finishing.

Research

Y. T. Sihvonen, senior research physicist, Physics and Instrumentation Department, GM Research Staff, before a student group of the University of Buffalo, Buffalo, New York, February 14; title: High-Sensitivity Capacitance Pick-Up for Heart Sounds.

Carl E. Bleil, senior research physicist, Physics and Instrumentation Department, GM Research Staff, before the Southwestern Section of the American Physical Society, Norman, Oklahoma, March 2; title: Evaporated Film of Magnetic Materials.

Before the Conference on Analytical Chemistry and Applied Spectroscopy, Pittsburgh, March 4 to 8, GM Research Staff Chemistry Department research chemist speakers: **R. L. Chance**; title: The Spectrophotometric Determination of Silicon in Zinc-Base Die Cast Alloys; **Antrim H. Jones**; title: The Photometric Determination of Boron in High-Temperature Alloys Using Quinalizarin; **R. E. Kohn**; title: The Determination of Tungsten and Molybdenum in Tool Steels Using Anion Exchange Separation; **Richard B. Loranger**; title: The Photometric Determination of Vanadium in Tungsten Steel by the Phosphotungstovanadate Method; **A. C. Ottolini**; title: Spectrochemical Determination of Bismuth and Magnesium in Cast Iron; and **Paul K. Winter,** research associate; title: An Electrogravimetric Method for the Determination of Copper in the Presence of Iron. GM Research Staff Physics and Instrumentation Department research physicist speakers: **R. F. Majkowski** and **T. P. Schreiber**; title: The Effects of Various Oxidizing Atmospheres on the Spark Excitation of a Nickel-Base Alloy; **T. P. Schreiber** and **D. L. Fry,** supervisor of Department; title: Spectral Character as a Function of Source Parameter.

Helen B. Bartlett, supervisor of ceramic research, Research Department, AC Spark Plug Division, before the Flint branch of the Michigan Society of Professional Engineers, March 18; title: Gems, Natural and Synthetic.

Before the American Physical Society, Philadelphia, March 21; **Donald Koistinen** and **R. E. Marburger,** research physicists, Physics and Instrumentation Department, GM Research Staff; title: Extent of the Austenite-Martensite Transformation in Pure Iron-Carbon Alloys; **R. E. Marburger** and **T. P. McKinney,** physics technician, Physics and Instrumentation Department; title: The Investigation of an Aluminum-Silicon-Cadmium Alloy by Electron and X-ray Diffraction; **R. E. Marburger** and **A. W. Schlucter,** research engineer, Mechanical Development Department; title: Precipitated Phases and their Distribution in Al-Si-Cd Alloy.

Robert J. Moffat, senior research engineer, Gas Turbines Department, GM Research Staff, before the A.S.M.E. gas turbine conference, Detroit, March 21; title: Designing Thermocouples for Response Rate.

Robert C. Frank, research physicist, Physics and Instrumentation Department, GM Research Staff, before the Indiana Spectrographers Society, Indianapolis, April 8; title: Principles and Applications of Mass Spectroscopy.

Before the American Ceramic Society, Dallas, Texas, May 6 to 9, AC Spark Plug Division Research Department participants: **Helen B. Bartlett,** supervisor of ceramic research; Basic Science Division session chairman; **C. F. Schaefer,** supervisor of ceramic development and control; session chairman of Symposium on Alumina Ceramics, title: Review of Forming Methods for High Alumina Ceramics; and **Karl Schwartzwalder,** director of research; presidential address to the Society.

Solution to the Previous General Motors Institute Laboratory Problem

When an automobile accelerates from rest a certain portion of the available horsepower is absorbed in accelerating the four wheels. Before calculating the horsepower required to accelerate the four wheels, the polar moment of inertia of the wheel must be determined. The solution presented here is unique in that a torsional pendulum was used to obtain the polar moment of inertia. The maximum horsepower required to accelerate the four wheels from rest to 60 mph in 10 seconds at a constant rate was found to be *11.44 hp*.

The first step in the solution is to determine the polar moment of inertia of the tire and wheel and mounting plate assembly (Fig. 1). This may be determined by using the following fundamental vibration formula:

$$T = \frac{2\pi}{\sqrt{\dfrac{K}{J_{wp}}}} \qquad (1)$$

where

T = natural period of the torsional pendulum system (sec)

K = torsional stiffness of the suspension wire (in-lb per radian)

J_{wp} = polar moment of inertia of the tire and wheel and mounting plate assembly (lb-in-sec^2).

Before the polar moment of inertia can be determined, however, it is necessary first to calculate the stiffness K of the steel suspension wire. The following formula applies for the calculation of the torsional rate of a wire of known length, diameter, and composition:

$$K = \frac{\pi D^4 G}{32L} \qquad (2)$$

where

D = wire diameter (in.)

G = modulus of rigidity of steel (psi)

L = length of the wire (in.).

Find the Horsepower to Give 4 Wheels of an Automobile a Certain Acceleration at a Specified Velocity

Faculty Member-in-Charge:
MERLE L. DEMOSS
G.M.I. Cooperative Students:
THEODORE N. LOUCKES
Oldsmobile Division
and ROBERT B. ROBINSON
Chevrolet Motor Division

Accelerating hp: a function of polar moment and wheel velocity as well as force

The problem stated that the diameter D of the steel suspension wire was 0.051 in., the length L was 81.875 in., and the modulus of rigidity G was 12×10^6 psi. Substituting these values in equation (2) gives the following value for stiffness K of the suspension wire:

$$K = \frac{\pi(0.051)^4 \ (12 \times 10^6)}{32(81.875)}$$

$$K = 0.0980 \text{ in-lb per radian.}$$

Solving equation (1) for the polar moment of inertia J_{wp} of the wheel and plate assembly gives:

$$J_{wp} = \frac{KT^2}{4\pi^2}. \qquad (3)$$

The natural period of oscillation T was stated in the problem as 61.81 seconds. Substituting this value and the computed value of torsional stiffness K into equation (3) gives the following value for the polar moment of inertia J_{wp} of the wheel and plate assembly:

$$J_{wp} = \frac{(0.0980) \ (61.81)^2}{4\pi^2} = 9.50 \text{ lb-in-sec}^2.$$

The polar moment of inertia of the mounting plate J_p may be calculated by the following formula:

$$J_p = M\left(\frac{l^2 + w^2}{12}\right) \qquad (4)$$

where

J_p = polar moment of inertia of the mounting plate (lb-in-sec^2)

M = mass of the mounting plate (lb-sec^2 per in.)

l = length of plate (in.)

w = width of plate (in.).

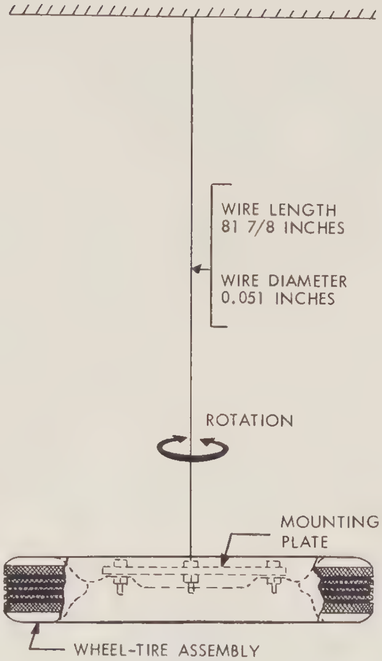


Fig. 1—The moment of inertia of a tire and wheel assembly can be determined by a torsional pendulum. The assembly is suspended from a mounting plate attached to a steel wire of known composition, diameter, and length. The wheel is turned about its axis of rotation and the natural period of oscillation observed.

Substituting in equation (4) the known values for the weight, length, and width of the mounting plate, as stated in the problem, gives the following value for the polar moment of inertia of the plate:

$$J_p = \left(\frac{2.55}{386}\right)\left(\frac{36 + 36}{12}\right)$$

$$J_p = 0.0397 \text{ lb-in-sec}^2.$$

This value of inertia is small in comparison to that of the wheel. It seems, however, that at this point it will affect a 3-figure slide rule accuracy.

Therefore,

$$J_w = 9.50 - 0.04 = 9.46 \text{ lb-in-sec}^2 \qquad (5)$$

where

J_w = polar moment of inertia of the wheel.

To calculate the power required to accelerate the wheel both in translation and rotation, it is convenient to determine the moment of inertia about the instant center at the surface of the road. This determination can be made by using the following equation:

$$J_c = J_w + M_w R^2 \qquad (6)$$

where

J_c = polar moment of inertia of the wheel about an instant center at the surface contact point (lb-in-sec^2)

M_w = mass of the wheel (lb-sec^2 per in.)

R = rolling radius of the wheel (in.).

Presented here is the solution to a typical problem initiated and solved by students enrolled in advanced laboratory course work at General Motors Institute. This particular problem was solved during a 4-hour Automotive Chassis—Design and Testing Laboratory period. The problem allowed the students to apply principles of physics and dynamics to the solution of a typical automotive design analysis.

The problem stated that there were 723.35 wheel revolutions per mile at a speed of 60 mph. From this the rolling radius of the wheel can be determined as follows:

$$R = \frac{5{,}280 \text{ ft per mile}}{(723.35 \text{ revolutions per mile})\ (2\pi)}$$

$$R = 1.163 \text{ ft}$$

$$R = 14.0 \text{ in. (rounding off to three figures).}$$

Substituting the value for the rolling radius R of the wheel in equation (6) gives the following value for the moment of inertia J_c about the instant center:

$$J_c = 9.46 + \frac{46.5}{386}\ (14.0^2)$$

$$J_c = 33.2 \text{ lb-in-sec}^2.$$

The above calculation indicates that the moment of inertia J_p of the plate could have been ignored.

The force required to accelerate one wheel may be obtained from the following work energy equation, which assumes there is no slipping and that the axis of the wheel is a center of gravity axis:

$$M_c\Theta = \frac{1}{2} J_c\omega^2 \qquad (7)$$

where

M_c = moment of the wheel about the instant center (lb-in.)

ω = maximum angular velocity (radians per sec)

Θ = angular displacement (radians).

Substituting in equation (7) the force F applied at the axis of the wheel times the rolling radius for the movement about the instant center gives:

$$FR\Theta = \frac{1}{2} J_c\omega^2\ . \qquad (8)$$

Equation (8) can be rearranged as follows:

$$F = \frac{J_c\omega^2}{2R\Theta}\ . \qquad (9)$$

The maximum angular velocity ω can be obtained from the following relationship:

$$V = R\omega$$

where

V = velocity (in. per sec).

Therefore,

$$\omega = \frac{V}{R} = \frac{(88)\ (12)}{14} = 75.5 \text{ radians per sec.}$$

The distance S traveled by the wheel can be expressed by the following fundamental formula:

$$S = \frac{1}{2} at^2 \qquad (10)$$

where

S = distance traveled (ft)

a = acceleration (ft per sec²)

t = elapsed time (sec).

Substituting the known values for a and t into equation (10) gives the following value for S:

$$S = \frac{8.8}{2}\ (10^2) = 440 \text{ ft.}$$

The angular displacement Θ can be expressed as:

$$\Theta = \frac{S}{R}$$

$$\Theta = \left(\frac{440}{14}\right) 12 = 377 \text{ radians.}$$

Substituting in equation (9) the known values for J_c, ω, R, and Θ, gives the following value for the force F applied at the axis of the wheel:

$$F = \frac{(33.2)\ (75.5)}{(2)\ (14)\ (377)} = 17.9 \text{ lb.}$$

The horsepower hp required to accelerate one wheel of the car may be expressed by the following equation:

$$hp = \frac{FV}{550}$$

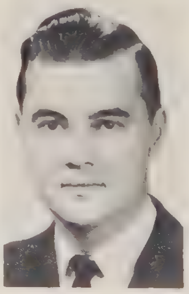
$$hp = \frac{(17.9)\ (88)}{550}$$

$$hp = 2.86.$$

Multiplying the horsepower required to accelerate one wheel by 4 gives 11.44 horsepower required to accelerate the 4 wheels from 0 mph to 60 mph at a constant rate of 8.8 ft per sec².

Student Contributors to the Typical General Motors Institute Laboratory Problem

THEODORE N. LOUCKES, is a detail engineer in the Engineering Department of Oldsmobile Division. He joined Oldsmobile in 1948 as a G.M.I. co-op student. His schooling was interrupted in 1951 when he joined the Air Force. He was separated from service in 1955 with the rank of first lieutenant after serving as a pilot. He returned to G.M.I. and completed the Four-Year cooperative program in February of this year. He is presently participating in the Fifth-Year program which involves an assigned project study dealing with the study of valve motion and the influence of valve train components on this motion. Mr. Louckes gained previous experience at Oldsmobile on the testing and evaluation of experimental automotive components for durability and functional operation.



ROBERT B. ROBINSON, is a junior engineer in the Product Engineering Department of Chevrolet-Flint Manufacturing. He is presently participating in the G.M.I. Fifth-Year Program which will qualify him for a baccalaureate degree in mechanical engineering when completed.

Mr. Robinson's current work as a junior engineer is concerned primarily with clutches—their testing and adaptability to manufacturing. During the Four-Year G.M.I. cooperative program his plant work assignments were in such areas as product testing, dynamometer testing, and stress and torsional vibration analysis.

Mr. Robinson was with the U.S. Army Signal Corps for two years and saw service in Korea.

Solution to the Previous Problem:

Determine the Design Specifications for a Pressure Regulator Coil Spring

By IVAN K. LUKEY
Buick Motor Division
and WILLIAM H. LICHTY
General Motors Institute

Spring material selected to meet stress, temperature, and space requirements

The use of a coil spring acting in conjunction with an oil pressure regulating valve in an automatic transmission presents a typical spring problem requiring experience and/or trial solutions before all spring specifications are established. In this particular problem, space limitations imposed by the valve body design require a coil spring capable of withstanding high stress. This stress, coupled with the temperature at which the spring operates and the load loss limitation, dictates the selection of a spring material having the required stress and heat resisting properties. The material selected, in this case a chrome-silicon alloy steel, must then be given a final check to see if it meets the critical limitation—solid height and stress at solid height. This is the solution to the problem presented in the April-May-June 1957 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

WHEN the automatic transmission operates in either Direct Drive or Reverse, the regulating valve must control an oil pressure required for each operating condition. The regulated pressure required for Direct Drive operation is 41 psi (Fig. 1a). The regulated pressure required for Reverse operation is 71 psi (Fig. 1b).

The coil spring, acting as the regulating force on the valve, has a Direct Drive operating length L_1 of 0.940 in., a Reverse operating length L_2 of 0.700 in., and is subjected to two forces F_1 and F_2 under each operating position (Fig. 2).

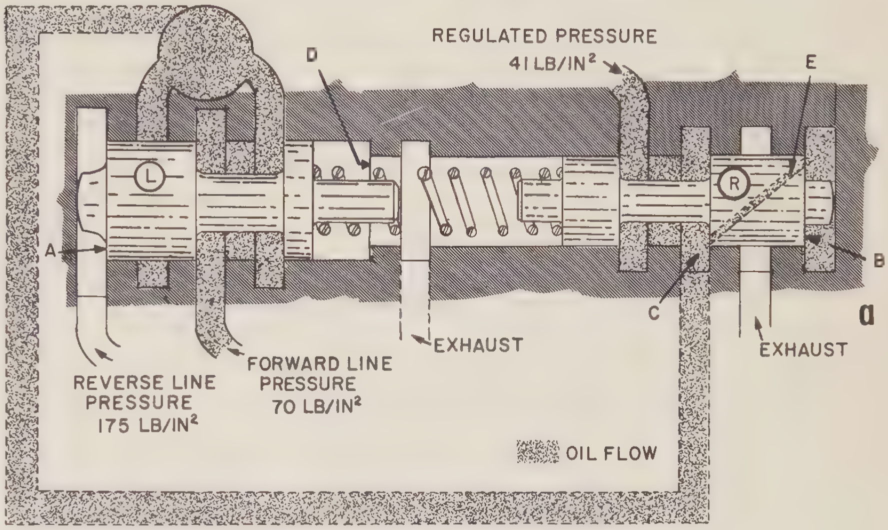
The diameter of the regulating valve is 0.375 in. The area of the valve (area B, Fig. 1) exposed to oil pressure, therefore, is equal to $(\pi)\ (0.375)^2/4$ or 0.1104 sq in. Forces F_1 and F_2 acting on the coil spring can be calculated as follows:

$$F_1 = 41(0.1104) = 4.5280 \text{ lb}$$
$$F_2 = 71(0.1104) = 7.8412 \text{ lb.}$$

Establishment of forces F_1 and F_2 acting on the coil spring while it is in operating lengths L_1 and L_2 allows the spring rate to be calculated as follows:

$$\text{Spring rate} = \frac{F_2 - F_1}{L_1 - L_2} = \frac{7.8412 - 4.5280}{0.940 - 0.700}$$

$$\text{Spring rate} = 13.8008 \text{ lb per in.}$$



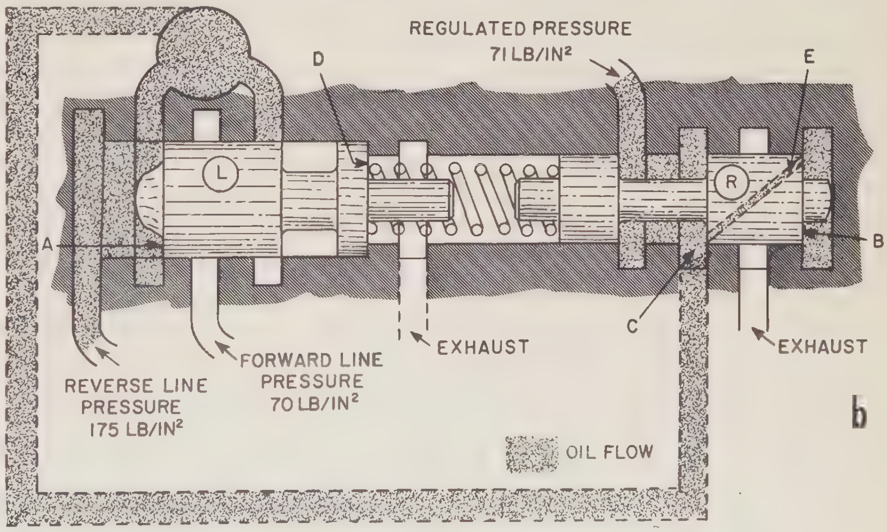


Fig. 1—When the automatic transmission is shifted to Direct Drive (a), the regulating valve R receives oil at a pressure of 70 psi. The oil enters at supply port C and then flows through the drilled passage E. This creates a pressure build-up behind area B, which represents the area of the regulating valve exposed to oil pressure. The valve then controls the pressure to the required 41 psi. When the transmission is shifted to Reverse (b), an oil pressure of 175 psi is brought to bear on the exposed area A of valve L. This pressure causes valve L to move to the right, with shoulder D providing a stop. When valve L moves to the right, the 70 psi supply line is closed and the 175 psi supply line is opened for oil flow to the regulating valve R. This valve receives oil in the same manner as for Direct Drive operation. The valve reduces the oil pressure from 175 psi and maintains a regulated pressure of 71 psi.

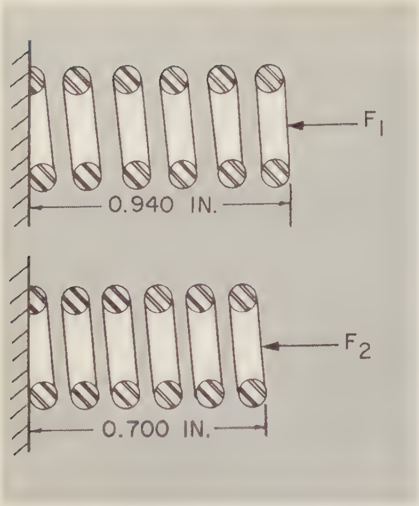


Fig. 2—The regulator coil spring is required to have an operating length of 0.940 in. when the transmission is in Direct Drive operation and an operating length of 0.700 in. when in Reverse operation. The spring, in turn, is subjected to force F_1 during Direct Drive and force F_2 during Reverse.

The spring must operate at a maximum oil temperature of 300° F with no more than three psi loss in regulated pressure. This amounts to a maximum load loss, or set, due to temperature of about four per cent. This requirement places some stringent limitations on the selection of a suitable spring material. A heat set data chart (Fig. 3) for three specific spring materials (music wire, oil-tempered spring wire, and chrome-silicon spring wire) indicates that music wire is unsuitable for this particular application. An oil-tempered spring wire material could be used if stresses were held to 70,000 psi. Preliminary calculations, however, will prove this to be impossible. It becomes necessary, therefore, to use a good alloy steel. A chrome-silicon spring wire material, which has high stress and heat resisting properties, is selected as the most suitable material. A working stress of 100,000 psi can be assumed as a starting point for calculations relating to spring wire diameter.

The wire diameter d can be calculated by using the following formula:

$$d = \left(\frac{8PD}{\pi S}\right)^{1/3} \qquad (1)$$

where

d = spring wire diameter (in.)

P = maximum working load (lb)

D = mean diameter of coil (in.)

S = working stress (psi).

The maximum working load P was calculated to be 7.8412 lb. The mean diameter D of the coil can be assumed to be 0.300 in. The working stress S is assumed to be 100,000 psi. Substituting these values into equation (1) and solving for d gives the following value for wire diameter:

$$d = \left[\frac{8(7.8412)\ 0.300}{3.1416(100,000)}\right]^{1/3} = 0.0391 \text{ in.}$$

The steel wire gage number (Washburn and Moen classification) which most closely approximates the calculated wire diameter is 19 gauge. The wire diameter corresponding to this gauge number is 0.0410 in.

Next, the *Wahl factor* must be determined. This is a corrective factor which takes into account the effect of the curvature of the wire on stress. The Wahl factor K is proportional to the mean coil diameter D and the wire diameter d according to the following relationship:

$$K = \frac{4C\text{-}1}{4C\text{-}4} + \frac{0.615}{C}$$

where

$$C = \text{spring index} = D/d.$$

The spring index C is equal to 0.300/0.041 or 7.3170. Using this value, the Wahl factor K can be calculated as follows:

$$K = \frac{4\ (7.3170)\ -\ 1}{4\ (7.3170)\ -\ 4} + \frac{0.615}{7.3170}$$

$$K = 1.2027.$$

The actual working stress S to which the spring will be subjected while under the maximum working load and operating length can now be determined as follows:

$$S = \frac{8PDK}{\pi d^3} \qquad (2)$$

Substituting the known values for P, D, K, and d into equation (2) will give the following value for the actual working stress:

$$S = \frac{8\ (7.8412)\ 0.300\ (1.2027)}{3.1416\ (0.041)^3}$$

$$S = 104,531 \text{ psi.}$$

This calculated value for the actual working stress, while above the initial stress estimate of 100,000 psi, is safely within the allowable working limits.

The deflection f per active coil can be calculated as follows:

$$f = \frac{8PD^3}{Gd^4} \qquad (3)$$

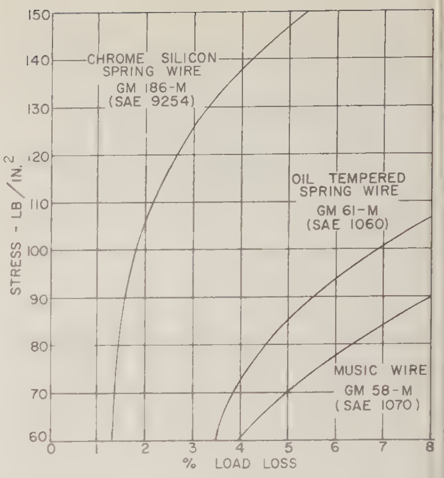


Fig. 3—This chart compares three different spring materials as to percentage of load loss due to heat set at 300° F for infinite time. The springs were stress relieved at 750° F for ½ hour after coiling. Data for this chart was obtained from the August 1952 and April 1953 issues of *The Mainspring*, published by the Associated Spring Corporation, Bristol, Connecticut. The GM numbers for each of the three spring materials specify not only chemical composition but also properties involving heat treatment, wire size, and other factors. The approximate equivalent of Society of Automotive Engineers chemical compositions for each of the GM materials are SAE 9254 (GM 186-M), SAE 1060 (GM 61-M), and SAE 1070 (GM 58-M).

where

f = deflection per active coil (in. per coil)

P = change in working load during compression, $F_2 - F_1$ (lb)

G = torsional modulus for the spring material (11,500,000 psi for chrome-silicon alloy steel)

D = mean coil diameter (in.)

d = wire diameter (in.).

Substituting the known values for P, D, G, and d into equation (3) and solving for f gives the following value for deflection per active coil:

$$f = \frac{8\ (3.3122)\ (0.300^3)}{11,500,000\ (0.410^4)}$$

$$f = 0.02201 \text{ in. per active coil.}$$

The number n of active coils required for the regulator spring will be equal to the deflection of the spring ($L_2 - L_1$) divided by the deflection f per active coil or

$$n = \frac{0.240}{0.02201} = 10.9041$$

$$n = 11 \text{ active coils.}$$

The solid height h of the spring is equal to the total number of coils N times the

wire diameter d. Adding one "dead" coil at each end of the spring for closing, the solid height h of the spring is:

$$h = Nd = (13)\ (0.041) = 0.5330 \text{ in.}$$

The deflection of the spring from its free length to its initial working position is found by dividing the load at the initial working position by the spring rate as follows:

$$\frac{F_1}{\text{spring rate}} = \frac{4.5280}{13.8008} = 0.32809 \text{ in.}$$

The total free length of the spring, therefore, is equal to its length at the initial working position (0.940 in.) plus the deflection between this position and its free length (0.32809 in.), or 1.2680 in.

A final check on the stress at solid height is needed. The full deflection possible is equal to the total free length minus the solid height, or 0.7350 in. The load at solid height is equal to the spring rate times the full deflection, or 10.1435 lb. The maximum stress S to which the spring will be subjected at solid height can be calculated as follows:

$$S = \frac{8PDK}{\pi d^3}$$

$$S = \frac{(8)\ (10.1435)\ (0.300)\ (1.2027)}{(3.14.6)\ (0.041)^3}$$

$$S = 135{,}225 \text{ psi.}$$

This calculated value for S is near the upper limit for maximum stress but still acceptable.

Summary

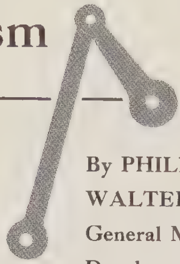
The complete specifications for the pressure regulator coil spring are as follows:

Material—chrome-silicon steel wire
Total number of coils—13
Total number of active coils—11
Wire diameter—0.041 in.
Mean coil diameter—0.300 in.
Free length—1.2680 in.
Solid height—0.533 in.

It should be noted that different spring materials or different combinations of variables could produce other satisfactory specifications. The specifications listed above, however, were used by the Transmission Department of Buick Motor Division in the design of a pressure regulator valve used in an automatic transmission.

A Typical Problem in Engineering:

Determine the Design Specifications for a Crank and Connecting Rod Transfer Mechanism



By PHILIP WEST and
WALTER D. NOON
General Motors Process
Development Staff

Assisted by Gerhard W. Sood
General Motors Institute

Solve a problem involving horsepower determination and force analysis

In recent years the nation's trend to greater production has brought about the increased use of specially designed automatic assembly machines. The problem of obtaining accurate, smooth, and reliable indexing action on small, in-line type assembly machines was satisfactorily solved by using properly designed, cam-controlled indexing, or transfer, mechanisms. As the in-line type of machine increased in size a new indexing mechanism was necessary to transfer heavier loads through a longer distance in a minimum of time. Such was the problem faced by engineers of the General Motors Process Development Staff (located at the GM Technical Center) when designing an in-line transfer machine to be used for the automatic assembly of automotive cylinder heads. A crank and connecting rod transfer mechanism appeared to have many desirable qualities for effecting smoothly the transfer of the cylinder heads. As part of the design procedure a detailed study of forces developed during the transfer and the power required to effect the transfer was necessary.

One of the most vital componenets of an automatic transfer assembly machine is its index, or transfer, mechanism. This mechanism must be designed to provide smooth, accurate, and reliable indexing action in a minimum of time.

Smoothness of index, in many cases, is important not only to eliminate shock loadings but, in assembly work, to keep loose parts in position between loading and fastening stations when they cannot be integrated. The movement of parts from work station to work station must be accurately synchronized so that adequate time is alloted for each automatic assembly operation to be performed. Indexing time in transfer machines is not generally productive time and must be kept to a minimum. Reliability and freedom from excessive maintenance is imperative, since downtime is expensive. These are the factors, then, that must be considered when designing a transfer mechanism for an automatic assembly machine. And these were the factors which had to be considered by engineers of the GM Process Development Staff preparatory to initiating the design of a transfer mechanism to move automotive cylinder heads while being assembled automatically.

Design Specifications and Considerations

The length of the in-line type of automatic assembly machine was to be 137 ft. The specifications called for each cylinder head to be moved a distance of 2 ft in a time of 1.6 seconds. The transfer mechanism was to move 62 cylinder heads, each weighing approximately 65 lb.

Because of the long index movement required, a cam-operated transfer mechanism as used on in-line type machines having a short index movement, would be impractical. The smoothness of indexing action which could be designed into a cam to effect the transfer had to be obtained some other way.

Design considerations indicated the desirability of having the cylinder heads ride on steel pallets. These pallets, guided on lubricated and hardened steel rails, would be connected to an endless chain. This would provide a means for returning

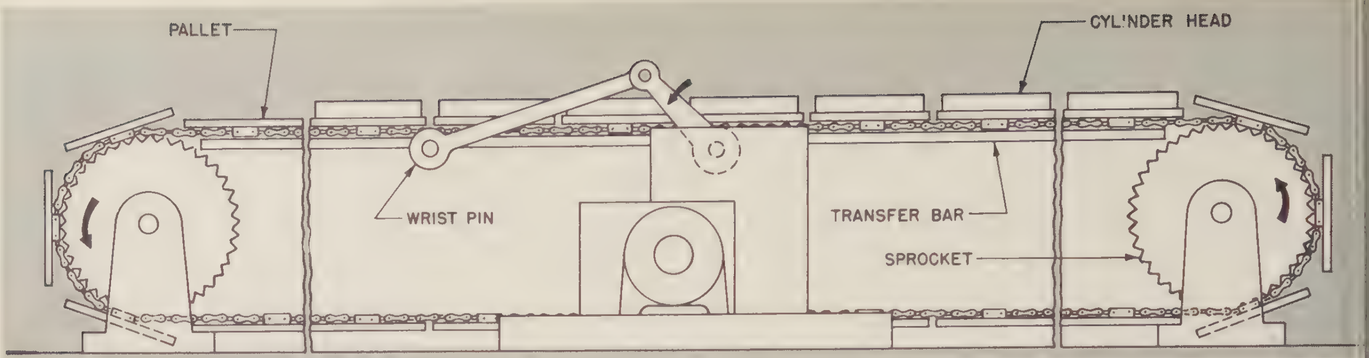


Fig. 1—A crank and connecting rod mechanism was selected as the means for transferring automotive cylinder heads through an automatic assembly process. The cylinder heads ride on steel pallets connected to an endless chain which passes over a sprocket located at each end of the 137 ft-long machine. Motion is imparted to the pallets by means of a transfer bar which rotates to engage or release driving lugs on the pallets. The transfer bar is moved linearly by a drive mechanism, which provides the means for accelerating and decelerating the load. Before any of the components of the system could be designed the maximum horsepower required at the crank and the maximum torque developed at the crank had to be calculated, along with the maximum force developed along the connecting rod. The maximum force at the wrist pin along the direction of travel and normal to the direction of travel also had to be calculated

the pallets underneath the machine without the need for an auxiliary drive or return mechanism. The endless chain would pass over two sprockets, one at each end of the machine.

Motion would be imparted to the pallets by means of a transfer bar. The transfer bar would rotate to engage or release driving lugs on the pallets. The bar would also have to be moved linearly by a drive mechanism. The drive mechanism would provide the means for accelerating and decelerating the load.

The total load to be indexed consisted of the transfer bar, pallets, chain, and sprockets, in addition to the cylinder heads. From a preliminary layout it was estimated that the total load to be indexed equaled 11,000 lb plus the two 25-in. diameter sprockets, each of which weighed 114 lb.

After considering several possible systems, a crank and connecting rod mechanism (Fig. 1) was selected as having many of the features desired for the driving mechanism. A complete cycle for the crank and connecting rod mechanism would be as follows:

(a) After the transfer bar rotated to engage the pallets, the crank would be rotated 180° and stopped
(b) The transfer bar would then be disengaged
(c) The crank would then rotate another 180° in the same direction and stop.

The geometry of the system provided a fixed acceleration and deceleration which could not get out of adjustment.

To evaluate fully the crank and connecting rod mechanism it was necessary to determine first the forces acting on the mechanism and the power required (Fig. 2). This information would then be used in designing the component parts.

A coefficient of friction of 0.2 was assumed for the system. Rotational friction of the sprockets and frictional effects of the downward thrust at the wrist pin were neglected, since this conservative value for the coefficient of friction would account for these factors.

Before the overall system could be analyzed, the length of the connecting rod had to be determined. Space and rigidity requirements dictated that the connecting rod should be as short as possible. However, a ratio of connecting rod to crank of less than approximately 3 to 1 would result in rapidly increasing accelerations and the formulation of loads normal to the direction of travel. The length of the connecting rod, therefore, was set at three times the length of the crank.

Since the total index time was set at 1.6 seconds, an allowance had to be made for relay and clutch operation time. This time was estimated at 0.2 seconds. It was necessary, therefore, to move the load the required distance in 1.4 seconds.

Problem

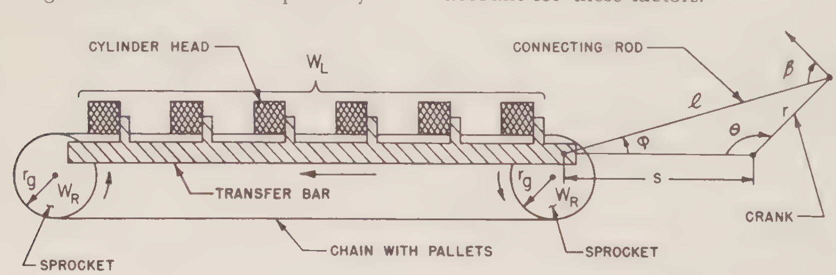
The problem is to determine the following design specifications:

- Maximum horsepower required at the crank
- Maximum torque developed at the crank
- Maximum force along the connecting rod
- Maximum force at the wrist pin
  - (a) along the direction of travel
  - (b) normal to the direction of travel.

A constant angular velocity for the crank is to be assumed.

Since the mathematical expressions for the load velocity and acceleration will be quite complicated, it will be practical to solve the problem graphically. A point should be plotted every 30° and at zero acceleration. In addition, the torque after 175° of rotation and the acceleration after 45° of rotation of the crank should be plotted.

The solution to the problem will appear in the October-November-December 1957 issue of the General Motors Engineering Journal.



ℓ = Length of connecting rod
r = Length of crank arm
S = Distance from wrist pin to center of crank rotation
r_g = Radius of sprocket
W_L = Total weight of heads, chain, and transfer bar
W_R = Weight of each uniform sprocket

Fig. 2—This schematic diagram illustrates the connecting rod and crank transfer mechanism.

A Typical Problem in Engineering:

Determine the Horizontal and Vertical Deflection of a Circular Beam

By ROBERT W. LEWIS
Allison Division
Assisted by John A. Straw
General Motors Institute

When an appropriate formula is not available the analytical approach must be used

A typical problem encountered many times in the analysis of jet engine and turbo-prop engine components is that of determining the deflection of curved members. Usually, straight beam deflection problems are solved with the aid of an appropriate formula found in an engineering handbook. Comparatively little information is available, however, on curved beam deflection analysis. The engineer in this case must resort to a thorough mathematical analysis.

To solve a problem in curved beam deflection requires writing an integral equation and then integrating over the length of the beam. The integral equations required, which may be obtained from curved beam theory[1], are as follows:

$$\Delta\Phi = \int_A^B \frac{M}{EI}\, ds \qquad (1)$$

$$\Delta y = \int_A^B \frac{M}{EI}\, x\, ds \qquad (2)$$

$$\Delta x = \int_A^B \frac{M}{EI}\, y\, ds \qquad (3)$$

where

$\Delta\Phi$ = angular deflection (radians)
Δy = vertical deflection (in.)
Δx = horizontal deflection (in.)
M = bending moment (lb-in.)
E = modulus of elasticity of the beam material (psi)
I = moment of inertia of the beam (in.^{4})
ds = an element of length along the neutral axis of the beam. In the case of a curved beam, the element of length is neither dx or dy
A and B = arc length coordinates on any two fixed points on the elastic curve of the beam.

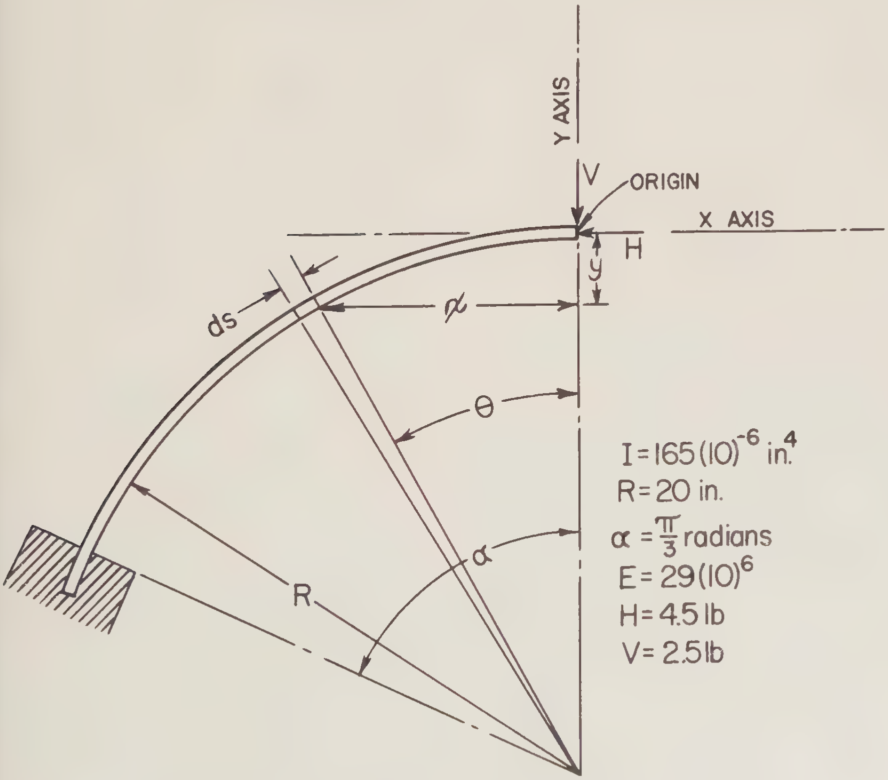


Fig. 1—When setting up the deflection equation for the curved beam of constant cross section shown here, the origin is taken at the free end of the beam.

It should be noted that equations (1), (2), and (3) are similar to the following more familiar area moment equations for straight beams:

$$\Delta\Theta = \int_A^B \frac{M}{EI}\, dx$$

$$\Delta y = \int_A^B \frac{M}{EI}\, x\, dx.$$

In equations (1), (2), and (3) no consideration is given to deflections due to axial compression or tension or deflections due to shear. In most cases these deflections are negligible, except for short, thick beams.

In a particular case where the curved beam is circular, it is very advantageous to convert to polar coordinates.

Problem

The problem is to determine the deflection of a circular beam of constant cross section (Fig. 1). The deflection is to be determined in both the vertical and horizontal direction. Assume that deflection due to stresses, other than flexure, is negligible.

In the solution to the problem equations (2) and (3) should be expressed in terms of R, Θ, and $d\Theta$ (Fig. 1). When setting up the equation for the deflection of the curved beam, the origin should be at the free end of the beam in all cases.

The solution to this problem will appear in the October-November-December 1957 issue of the General Motors Engineering Journal.

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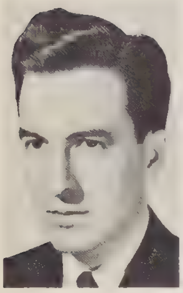

CARTER T. BRAGDON,

co-contributor of "Properties of Integral Seal Ball Bearings and Their Use by Engineers," is an automotive application engineer in the Product Engineering Department of New Departure Division. In this capacity he is responsible for the application of ball bearings—their lubrication, seals, and mountings for automotive products.

Mr. Bragdon joined New Departure in 1952 as a research and development engineer. His early projects were concerned with conducting bearing evaluation tests and investigating new materials for bearings. Many of the investigations performed concerned seal studies.

In 1953 Mr. Bragdon was appointed supervisor of seal development in the Research and Development Laboratory. In this position he supervised the design and testing of all new bearing seal designs and materials and contributed much to the development of seal evaluation test machines and in setting up evaluation test procedures. A primary aspect of his work was the development of the new removable synthetic rubber seal, of which he writes. Mr. Bragdon also performed liaison duty between the Engineering and Manufacturing Departments of New Departure in connection with this seal.

Mr. Bragdon was assigned as a general application engineer in 1955. The following year he assumed his present position.

Mr. Bragdon received a bachelor of science degree in mechanical engineering in 1952 from the University of Maine.

THEODORE L. CHISHOLM,

contributor of "Why Can't the English . . ." and coordinator of this issue's "Notes About Inventions and Inventors," is a patent attorney with the General Motors Central Office Patent Section where he is engaged in prosecution of patent applications covering devices invented by General Motors engineers. At present, his work is concerned primarily with inventions relating to automatic transmissions.

Mr. Chisholm's educational background includes studies in engineering and law. He was awarded the Bachelor of Engineering degree by Johns Hopkins University in 1919. He earned the Bachelor of Laws degree from George Washington University in 1926. While studying for the latter degree, he joined (in 1925) the General Motors patent office in Washington, D. C. He was associated with this office serving as patent attorney until 1932, when he left GM. In 1951 he returned to General Motors joining the Detroit office of the Patent Section. During the interim period he was engaged in patent work for Talon, Incorporated, Meadville, Pennsylvania; U. S. Rubber Company, New York, New York; and Johnson and Johnson, New Brunswick, New Jersey.

Mr. Chisholm is a veteran of World War I having served in the Signal Corps. His work has resulted in four patents granted in the field of refrigeration.

FRED H. COWIN,

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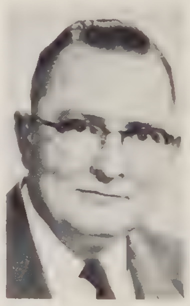
Mr. Cowin joined Cadillac in 1941 as a laboratory technician. In 1946 he became a project engineer. As such, he was concerned with developmental work on brakes and ride engineering analyses. Four years later he was promoted to assistant staff engineer. In this capacity he did supervisory work on car development and also engineering design and developmental work on transmissions, brakes, suspension, and steering. He assumed his present position in 1955.

The present engineering duties and responsibilities of Mr. Cowin as a staff engineer deal with the design and development of frames, suspension, steering, and wheels and tires. His major projects as a staff engineer are concerned with the Cadillac tubular center X-frame, low profile tires, and air suspension, of which he writes.

Mr. Cowin attended General Motors Institute as a co-op student sponsored by Fisher Body Division. After graduation from G.M.I. in 1939 he worked as a standards clerk at Fisher Body before joining Cadillac.

The technical affiliations of Mr. Cowin include the Society of Automotive Engineers.



MERLE L. DEMOSS,

faculty member-in-charge for the typical General Motors Institute laboratory problem "Find the Horsepower to Give 4 Wheels of an Automobile a Certain Acceleration at a Specified Velocity" and the solution appearing in this issue, is head of the Engineering Mechanics Section of the Department of Mathematics and Engineering Mechanics of G.M.I. Mr. DeMoss, who assumed his present position in 1955, also is a supervisor of Fifth-Year Project studies.

Mr. DeMoss received the B.S. degree from Northwestern State College, Alva, Oklahoma in 1938, majoring in mathematics and physics. He then went to Kansas State Teachers College as a Fellow in the Mathematics Department. He received his master's degree from this institution in 1939.

Mr. DeMoss was a member of the Mathematics Department faculty of the University of Kansas from 1939 to 1942, when he then joined the faculty of G.M.I. Shortly after joining G.M.I., Mr. DeMoss entered the service and was with the U. S. Navy from 1943 to 1946, when he then returned to G.M.I.

The technical affiliations of Mr.

DeMoss include the Mathematical Association of America, the Society for Experimental Stress Analysis, and the American Society for Engineering Education. He also is a member of the GM Master Mechanics Balancing Equipment Sub-Committee.

JOHN DOLZA,

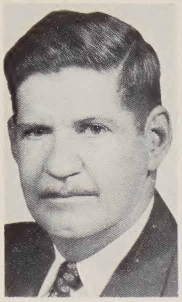
contributor of "A Discussion of the Basic Design and Operation of the General Motors Fuel Injection System," is engineer in charge of the Power Development Section of General Motors Engineering Staff. In this capacity Mr. Dolza supervises his group in contributing various designs of engine components to the General Motors car Divisions as well as developmental work on complete engines for the Armed Forces and for future consideration by GM Divisions. His activities also include developmental work on automotive air and refrigeration compressors.

Mr. Dolza's career with General Motors began in 1927 when he was employed as a draftsman for Buick Motor Division. Successive promotions led him to the position of assistant chief engineer. In 1940 Mr. Dolza was transferred to Allison Division where he was engaged as consulting engineer on special assignments. In 1945 he was transferred to the Power Development Section of the Engineering Staff and assumed his present position.

Mr. Dolza's previous major projects include developmental work on automatic controls for aircraft engines, aircraft turbo automatic controls, pilot automatic controls, lubrication systems for high-altitude aircraft engines, and turbo-prop engines and their related controls.

Mr. Dolza received a master's degree in both electrical engineering and mechanical engineering from the Polytechnic Institute, Turin, Italy, in 1926. While at the Institute, he designed the school's Altitude Laboratory.

Mr. Dolza's technical affiliations include membership in the Society of Automotive Engineers. He received the S.A.E.'s Manley Award in 1942. His work has resulted in the grant of 22 patents.



LAWRENCE J. KEHOE, JR.

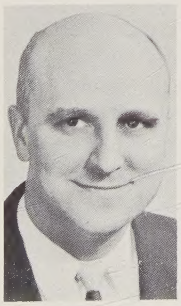
co-contributor of "The Development of the General Motors Air Spring," is administrative engineer of the Structure and Suspension Development Group of the GM Engineering Staff.

Mr. Kehoe joined General Motors in 1927 as a General Motors Institute co-op student sponsored by the Cadillac Motor Car Division. After graduation from G.M.I. in 1931 he worked with the Special Problems Group of Cadillac's Engineering Department. His work with this Group involved study of engine mountings, exhaust systems, and high-speed handling problems.

In 1940 Mr. Kehoe was transferred to the Central Office Engineering Staff's Product Study Group, which later became the present Structure and Suspension Development Group. Subsequent promotions as project engineer, senior engineer, and experimental engineer led to his present position, which he assumed in January 1957.

Prior to World War II, Mr. Kehoe did developmental work on ball-joint front suspensions and torsion bar suspensions. During World War II he did developmental work on the T-20 tank, which was later modified and became the M-46 tank. Before assuming his present position, Mr. Kehoe was engaged in the design and development of single-leaf springs and the air spring, of which he writes.

Mr. Kehoe is a member of the Society of Automotive Engineers and the GM Structure and Suspension Sub-Committee of the General Technical Committee.



WILLIAM H. LICHTY,

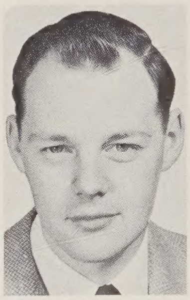
co-contributor of the problem "Determine the Design Specifications for a Pressure Regulator Coil Spring" and the solution appearing in this issue, is a member of the Product Engineering Department of General Motors Institute. He is in charge of automotive chassis and transmission design courses, a position he assumed in 1954.

Mr. Lichty joined General Motors in 1940 as a co-op student sponsored by the Cadillac Motor Car Division. After graduation from G.M.I. in 1944, with a diploma in automotive engineering, he worked in the Cadillac engineering laboratory. In 1947 he left Cadillac to continue his education.

Mr. Lichty received the B.S. degree in mechanical engineering from the University of Michigan in 1948. After receiving his master's degree the following year, he joined the faculty of G.M.I.

In addition to the automotive design courses, Mr. Lichty's teaching experience includes such mechanical engineering subjects as thermodynamics, fluid mechanics, and machine design. He is a supervisor of Fifth-Year Project studies.

The technical affiliations of Mr. Lichty include the Society of Automotive Engineers and the American Society for Engineering Education. He is on the governing board of the mid-Michigan section of the S.A.E.



IVAN K. LUKEY,

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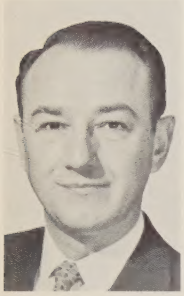
Mr. Lukey joined General Motors in 1952 as a General Motors Institute co-op student sponsored by Buick. In 1956 he completed the Four-Year cooperative program and is presently participating in the Fifth-Year program which, when completed, will qualify him for a bachelor's degree in mechanical engineering.

As part of the requirements for his degree, Mr. Lukey is currently working on an assigned project study involving the development of a practical method for calculating overall car performance with the aid of an IBM 705 digital computer. Included in the study is the development of the computer programming method and the application of numerical

methods for solving differential equations. In addition to his assigned project study, Mr. Lukey is also engaged in experimental engineering follow-up work dealing with new transmission designs.

The technical affiliations of Mr. Lukey include the Society of Automotive Engineers.



MICHAEL T. MONICH,

co-contributor of "Properties of Integral Seal Ball Bearings and Their Use by Engineers," is supervisor of standards and technical literature in the Product Engineering Department of New Departure Division. His responsibilities include the establishment and maintenance of product standards and specifications, aiding in the establishment of engineering administrative procedures, preparation of technical literature, and representing New Departure in various ball bearing standardization programs.

Mr. Monich joined the Engineering Department of New Departure in 1941 after receiving the B.M.E. degree from Pratt Institute. His early assignments were associated with product development and improvement and he contributed to various seal and separator developments which were adopted for military ball bearing applications. During this period he also was on the faculty of the University of Connecticut as an evening instructor and taught classes in machine design and organized classes in materials and processes.

In 1947 Mr. Monich was transferred to the Bearing Application Section of the Engineering Department. He served first as textile project engineer and then as general project engineer. In the latter capacity he supervised a group of project engineers specializing in various application fields. It was in this capacity that Mr. Monich gained firsthand experience in the capabilities and limitations of integral seal ball bearings, of which he writes.

Mr. Monich assumed his present position in 1955. His technical affiliations include committee membership in the American Standards Association and the Society of Automotive Engineers.

VON D. POLHEMUS,

co-contributor of "The Development of the General Motors Air Spring," is engineer in charge of the Structure and Suspension Development Group of the General Motors Engineering Staff.

Mr. Polhemus began his career with General Motors in 1933, when he joined Cadillac Motor Car Division after receiving a B.S.M.E. degree from the University of Cincinnati. He worked first as a laboratory assistant on the study of engine friction and, later, on transmission development.

In 1937 Mr. Polhemus became a senior project engineer with the Product Studies No. 2 Group of the Central Office Engineering Staff. In 1947 he was promoted to assistant engineer in charge of this Group, which later became the present Structure and Suspension Development Group. He assumed his present position in 1951.

Mr. Polhemus' Group is responsible for the development of new and improved types of passenger car structures and suspension systems. Many of the designs resulting from the developmental projects have been put into production, in part or total, by the GM car Divisions as, for example, the air spring by Cadillac.

Mr. Polhemus is a member of the Society of Automotive Engineers and the Engineering Society of Detroit. He is a member of the S.A.E. Riding Comfort Sub-Committee and also is chairman of the Structure and Suspension Sub-Committee of the General Technical Committee of General Motors.

CHARLES L. TUTT, JR.,

contributor of "Preparation and Evaluation of an Industrial Report," is Administrative Chairman of the Fifth-Year and Thesis Programs at General Motors Institute. He joined the staff of G.M.I. in 1946 as assistant to the director and assumed his present position in 1950.

Mr. Tutt received the B.S. degree in engineering in 1933 and the degree of mechanical engineer in 1934 from Princeton University. He then joined Buick Motor Division as a student engineer. In 1936 he was promoted to an engineer in the Chassis Unit Section and two years later was made a special assignment engineer.

In 1940 he joined the faculty of Princeton as assistant professor of mechanical engineering. While at Princeton, Mr. Tutt also served as a staff assistant for the American Society of Mechanical Engineers and acted as a consultant on projects for the Army Air Corps, Civil Aeronautics Administration, U. S. Navy Bureau of Yards and Docks, and the Chambersburg Engineering Company.

In 1944 Mr. Tutt obtained a leave of absence from Princeton to work for the McGraw-Hill Publishing Company as associate editor of *Product Engineering*. Two years later he joined the staff of G.M.I.

Mr. Tutt is an active member of the A.S.M.E. His other technical affiliations include the American Society for Engineering Education, the American Society of Tool Engineers, and the Society of Automotive Engineers. Mr. Tutt also is a member of the Sigma Xi and Alpha Tau Iota honorary societies and is a registered professional engineer in the state of New Jersey.

The General Motors Engineering Journal is a publication designed primarily for use by college and university educators in the fields of engineering and the sciences. Educators in these categories may, upon request, be placed on the mailing list to receive copies regularly. Classroom quantities also can be supplied regularly or for special purposes.

An index of Volume 4 of the Journal will be published in the next issue, October-November-December, 1957. Similarly, an index of each of the previous volumes has been published in the final issue of each volume. Additional copies of these index issues, as well as certain other back issues, are available to educators upon request.

Faith of the Engineer

I AM AN ENGINEER. In my profession I take deep pride, but without vainglory; to it I owe solemn obligations that I am eager to fulfill.

As an Engineer, I will participate in none but honest enterprise. To him that has engaged my services, as employer or client, I will give the utmost of performance and fidelity.

When needed, my skill and knowledge shall be given without reservation for the public good. From special capacity springs the obligation to use it well in the service of humanity; and I accept the challenge that this implies.

Jealous of the high repute of my calling, I will strive to protect the interests and the good name of any engineer that I know to be deserving; but I will not shrink, should duty dictate, from disclosing the truth regarding anyone that, by unscrupulous act, has shown himself unworthy of the profession.

Since the Age of Stone, human progress has been conditioned by the genius of my professional forebears. By them have been rendered usable to mankind Nature's vast resources of material and energy. By them have been vitalized and turned to practical account the principles of science and the revelations of technology. Except for this heritage of accumulated experience, my efforts would be feeble. I dedicate myself to the dissemination of engineering knowledge, and especially to the instruction of younger members of my profession in all its arts and traditions.

To my fellows I pledge, in the same full measure I ask of them, integrity and fair dealing, tolerance and respect, and devotion to the standards and the dignity of our profession; with the consciousness, always, that our special expertness carries with it the obligation to serve humanity with complete sincerity.

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